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Solar thermal air conditioning technology reducing the footprint of solar thermal air conditioning

Naci Kalkan [a],*, E.A. Young [a], Ahmet Celiktas [b]

[a] University of Southampton, School of Engineering Sciences, MSc Sustainable Energy Technologies, University Road, Southampton, SO17 1BJ, UK
[b] University of Southampton, School of Chemistry, Southampton, SO17 1BJ, UK

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A B S T R A C T

This paper describes heat driven cooling technologies in combination with solar thermal energy. A short overview about solar refrigeration systems is explained with a basic analysis of thermodynamic. Furthermore, new developments of open (desiccant cooling) and closed (absorption and adsorption) cooling cycles are presented and some of the new technologies are demonstrated in more detail. Additionally, recent installations of solar-thermal of air conditioning systems are described as examples with their working performance and system description. This report also includes small scale solar thermal absorption cooling system design in the following pages. The general purpose of the design is to understand how efficiently solar cooling system generates cooling, and to reduce the footprint of systems for integration with existing and future domestic buildings.

Contents

* Corresponding author. Tel.: +44 7550465437.
E-mail address: naci.kalkan@hotmail.com (N. Kalkan).

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1. Introduction

In the global world, our energy demand is produced from sources primarily from fossil fuels (coal, oil and natural gas etc.) that are non-renewable and limited in supply and sources from alternative energy that are making progress such as solar energy, wind power and moving water. Some estimates claim that our crude oil and natural gas reserves will be depleted within next 50 years; however, another problematic issue is the side effects of using fossil fuels that combustion of them is considered to be the number one factor for the release of greenhouse gases. Furthermore, population growth is another very important factor as it will almost reach nine billion people over the next 50 years. The world's energy demands will increase proportionately, energy shortage associated with environmental issues will be important for replacing fossil fuel energy production with renewable clean energy supply [1].

Increasing living standards and demand for human comfort has caused an increase in energy consumption. According to the International Institute of Refrigeration in Paris, the amount of electricity production from different types of refrigeration and air conditioning process is approximately 15% of all the electricity produced in the world. On the other hand, electricity consumption for air conditioning systems has been estimated around 45% of the whole residential and commercial buildings. For example, vapour compression based refrigeration systems consume much electricity and lead to the reduction of the valuable fossil fuel sources and production of the greenhouse gases that cause ozone layer depletion [2]. Furthermore, some refrigerants such as CFCs, HCFCs and HFCs, which result in ozone layer depletion, have been prohibited by the Montreal and Kyoto Protocol.

In recent years solar energy for environmental control has received much more attention in the engineering fields, as a result of the world energy shortage [1]. Particularly, summer air conditioning solar systems have been a growing market for both residential and commercial buildings. The average annual daily temperature of UK is expected to rise between 2 and 4 °C by 2080. This is a period that population growth goes with increased usage of domestic air-conditioning during summer while industrial usage reduced [3]. The reports on the demand for air-condition usage in Europe in 2000 showed that air-conditioned floor space had increased from 30 million m^2 in 1980 to over 150 million m^2 in 2000 [4].

Solar energy might be used for air conditioning (cooling systems) in two methods; photovoltaic solar cooling (conventional air conditioned based) and heat driven sorption system. The initial cost for solar photovoltaic cell is very high because the development of photovoltaic cell is very slow. Although different heat driven cooling technologies are available on the market, most of them have more than 50 kW capacities. In addition to this, the main issues comprise the large scale application, high initial cost, control and operation of these systems. Moreover, small scale solar thermal cooling systems have not been available on the market for a long time. Nevertheless, recently, some companies have begun to improve water chillers in the power range less than 50 down to 5 kW and first commercial

Nomenclature

A	Absorber area
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
c_0	Optical efficiency
c_1	Loss coefficient
c_2	Quadratic loss coefficient
CFC	Chlorofluorocarbons
COP	Coefficient of performance
DCS	Desiccant cooling systems
ECOS	Evaporative cooled sorptive
G	Incident global solar radiation on the collector surface
GAX	Generator/absorber heat exchanger
GCHP	Ground-coupled heat pump
GHE	Ground heat exchanger
GSHP	Ground-source heat pump
GWHP	Ground water source heat pump
HCFC	Hydrochlorofluorocarbons
HDPE	High-density polyethylene
HFC	Hydrofluorocarbons
IEA	International Energy Agency
MODESTORE	Modular high energy density sorption storage
Q_e	Cooling power
Q_g	Power supply
Q_{gen}	Heat input to the generator
Q_H	Heat added into the heater
Q_L	Heat transferred from the low temperature space
$\dot{Q}_{loss,\ conductive}$	Heat conduction losses
$\dot{Q}_{loss,\ convective}$	Connective losses
$\dot{Q}_{loss,\ opt}$	Optical losses
$\dot{Q}_{use}$	Useful heating power of the collector
SF	Solar fraction
SWHP	Surface water heat pump
T_{amb}	Ambient temperature
T_{av}	Average fluid temperature in the collector
$T_{c,o}$	Machine outlet cooling fluid temperature
$T_{g,i}$	Generator inlet temperature
$T_{t,o}$	Evaporative cooling tower outlet re-cooling fluid temperature
$W_{net,in}$	Net work input to the chiller
W_{pump}	Work input to the pump
η	Collector efficiency

systems are already available. Further development of small capacity cooling and air conditioning systems are still in high interest.

The focus of this report is to investigate and design solar refrigeration system particularly small scale solar thermal air conditioning systems, demonstrating that these systems' structures with new developments and several examples which have been studied and carried out previously. In order to reduce the footprint and increase the performance of solar thermal air conditioning system, small scale and highly efficient sub-system components are considered for the design. For instance, only 4.5 kW absorption chiller is used to produce chilled water by integration of a 6 kW fan coil unit for the system to circulate chilled water from the cold store, providing comfort cooling to the building. Other main components will be described at the design section. Presently, Solar Heating and Cooling Programme of the International Energy Agency (IEA) have started to carry out several research and demonstration projects in many countries and also in international co-operative projects. Also, one of the most important projects is the solar air conditioning in Europe that was set up in early 2002 and was managed over the next 2 years by a group of researchers from five countries, supported by European Commission.

Heat-driven cooling technologies involve mainly in closed cycles (absorption and adsorption) and open cycles (desiccant systems) [5,6]. More detailed description of these cycles is given in Section 2. Basically, these solar cooling systems contain solar thermal collectors which are connected to thermally driven cooling mechanisms. These systems consist of several components (Fig. 1): the heat driven system, the air conditioning system, heat-driven cooling device, solar collectors, a heat buffer storage, a cold storage and auxiliary subsystem.

2. Solar refrigeration systems

Solar refrigeration systems can be divided into two categories as electricity driven and thermal driven systems. This study was based on absorption refrigeration that is a sub-category of solar thermal driven systems; however, the other cooling technologies were introduced briefly by starting from electricity driven refrigeration systems. In what follows, thermal driven refrigeration systems were described in detail.

2.1. Electricity driven solar refrigeration systems

2.1.1. Vapour compression refrigeration systems

On the market for air conditioning and refrigeration, vapour compression systems are the most widely used method [8]. A vapour compression refrigeration system uses a circulating liquid refrigerant sealed in the system and it is circulated through various components of the system. The refrigerant absorbs and removes heat from the space to be cooled, afterwards rejects that heat elsewhere while passing through the components in the system. Process is the name of the change in the state of vapour whereas repetition of a series of similar processes is called a cycle.

The carnot cycle is one of the fundamental refrigeration cycles with the reversibility of all processes taking part in this cycle where heat would be converted to the mechanical work and vice versa. The efficiency of refrigeration cycle is directly proportional to difference between the highest and lowest temperature reached in one cycle.

The Rankine cycle converts heat into work most commonly in power generation, though energy losses by condensed vapour stream during the cycle. The mechanical power compression continuously recycles this energy by reversing the Rankine cycle. The conversion of the Rankine cycle system into mechanical power compression requires replacement of the turbine in the vapour compression cycle with an expansion valve or a capillary tube. If the turbine remains in the cycle, system becomes more complicated and expensive. The coefficient of performance (COP) increase of the system will be in a very small amount.

A schematic of the vapour compression cycle is given in Fig. 2. During the first part of the vapour compression cycle, refrigeration fluid remains in the vapour phase, on the other hand during the subsequent part of the cycle remains in the liquid phase [9].

As seen from Fig. 2, there are four processes in the ideal vapour compression cycle:

1–2 Isentropic compression in a compressor; maximizes the network output without an increase or decrease in the entropy.

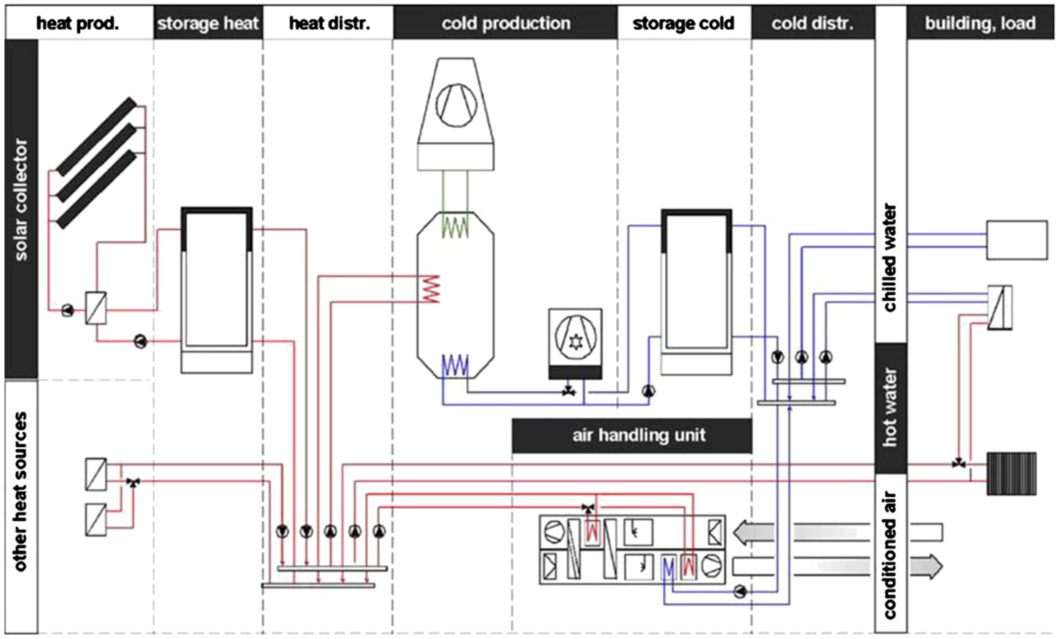

Fig. 1. Schematic description of solar-air conditioning system [7].

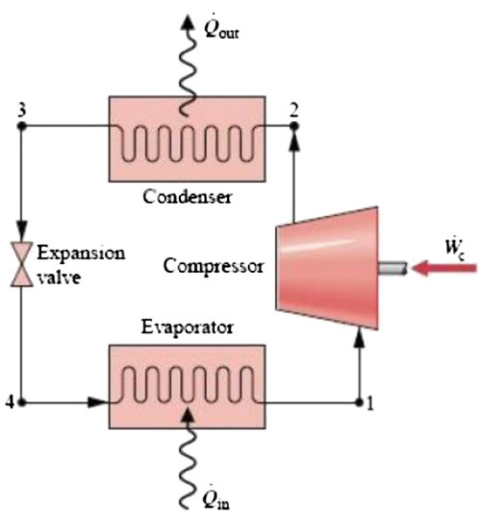

Fig. 2. Vapour compression cycle [10].

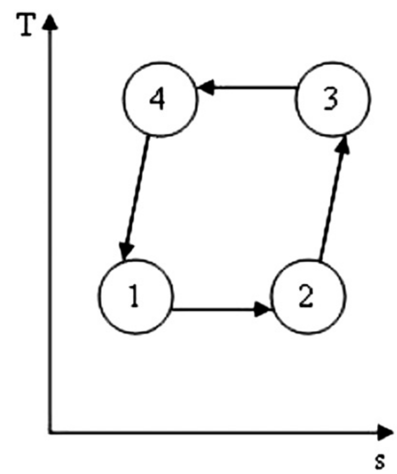

Fig. 3. Stirling cycle [13].

2–3 Constant pressure heat rejection to obtain a dry saturated liquid in a condenser.
3–4 Decreased pressure of refrigerant passes through the expansion valve.
4–1 Constant pressure heat absorption to obtain a dry saturated vapour in an evaporator.

COP is the efficiency of a refrigerator space explained as the ratio of energy seeks from the refrigerated space to the work input. According to Fig. 2;

$$\mathrm{COP} = \frac{Q_{in}}{W_c} \tag{2.1}$$

Typical COPs for vapour compression cycles are around 3 [11].

2.1.2. Thermoelectric refrigeration systems

If heat energy pumps out electronically from an insulated chamber in order to reduce the temperature by conversion of the temperature to electrical energy according to the Peltier–Seebeck effect. Thermoelectric cooling system has advantages such as being compact, being insensitive to motion or tilting, having a small 12-V fan as the only moving part that can operate directly from 12-volt batteries or electricity supplied from PV panels without any conversion, having longer life time, being totally environmental-friendly. The low COP near 0.1 that is suitable for lower cooling demands (under 25 W) is the main disadvantage of thermoelectric cooling [12].

2.1.3. Stirling refrigeration systems

Cyclic compression and expansion of air or other gas in a piston cylinder arrangement that compressed in the colder portion of the cylinder and expanded in the hotter portion of the cylinder is the basic principle for Stirling coolers. A temperature entropy diagram (*T*–*s* diagram) of the Stirling cycle is shown in Fig. 3. Stirling cycle consists of four thermodynamic processes as for the vapor compression cycle [13]:

1–2 Isothermal expansion.
2–3 Isochoric displacement.

3–4 Isothermal compression.
4–1 Isochoric displacement.

At moderate temperature differences between the heat sink and heat source, Stirling coolers have COPs close to vapour compression cycles, and perform better than vapor compression cycles if the temperature difference increases between sink and source [8]. Moreover, stirling coolers can provide very low temperatures as 10 K [8,13].

2.2. Solar thermal driven refrigeration systems

Cooling systems based on thermal energy generally have solar energy as the most widely available heat source for solar thermal driven cooling applications where a low temperature (below 200 °C) and/or cost efficient heat source is available. Industrial processes also produce waste heat that can be an alternative to the solar energy. On the market, four major solar thermal driven cooling systems are available with absorption, adsorption, desiccant and ejector cooling systems.

Furthermore, the Rankine power cycle for refrigeration can be driven by solar thermal energy to produce work output.

2.2.1. Absorption refrigeration

International Energy Agency (IEA) has presented the absorption cooling system as the most widely used solar thermal driven cooling system [14]. The idea for absorption cooling starts in the 1700s, and Ferdinand Carre got the first patent in 1859 with the ammonia–water refrigeration system [15]. Replacement of the compressor built in vapour compression cycle with a thermally driven absorption mechanism is the main difference of the absorption refrigeration cycle. An absorber, pump, expansion valve, regenerator and generator are the main parts of the absorption mechanism. The components of the absorption mechanism may change with the used working fluid in the system.

Usually water–ammonia (water/NH3) or lithium bromide–water (LiBr/water) solutions are used as the working fluid in absorption refrigeration cycles.

Water/NH3 systems employ water as the absorbent whereas NH3 is the refrigerant. In spite of that LiBr is the absorbent and water is the refrigerant in LiBr/water systems. Since the freezing point of NH3 is −77 °C, water/NH3 systems are feasible for low temperature applications. On the other hand, LiBr/water systems are only applicable for air conditioning applications with 0 °C freezing point of water.

Fig. 4 shows a simple single effect water/NH3 absorption refrigeration cycle. Apart from the components mentioned above, a rectifier is used in the absorption cooling system for this configuration. The rectifier provides separation of water vapour from the NH3 since water is highly volatile. The water vapour would freeze and accumulate in the evaporator without a rectifier; hence the system performance would decrease. The pump is the only part that needs work input to increase the pressure of a liquid in the absorption refrigeration cycle. And this work input is very small compared to the compressor in the vapour compression cycle system [9].

The COP of the absorption refrigeration cycle is defined as;

$$COP = \frac{Q_L}{Q_{gen} + W_{pump}} \cong \frac{Q_L}{Q_{gen}} \tag{2.2}$$

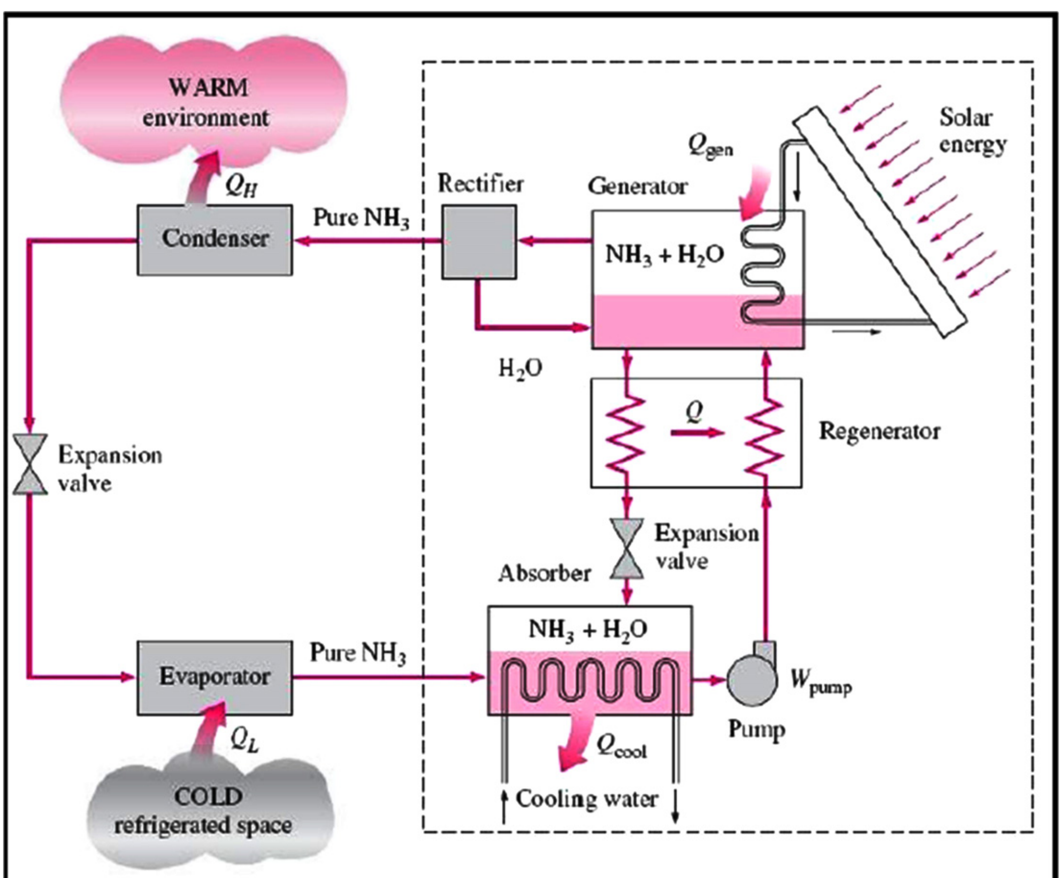

Fig. 4. Mechanism of a simple single effect water/NH_3 absorption refrigeration cycle [9].

where Q_L is the heat transferred from the low temperature space (refrigerated space), Q_{gen} is the heat input to the generator and W_{pump} is the work input to the pump which is neglected.

For a single effect absorption refrigeration cycle the COP is approximately 0.6 that is low compared to a vapour compression cycle. Since the vapour compression system powered with grid electricity, solar energy use for Q_{gen} supply will be make the absorption refrigeration system cost effective in term of the operational cost. However, when considered, the cost for investment absorption refrigeration system is more expensive than the vapour compression system, and absorption chillers are heavier and bulkier than the conventional vapour compression chillers. Commercially available double-effect refrigeration cycles are a member of multi-effect absorption cycle that increases COPs ranging from 0.8 to 1.2 [16]. The approaching triple-effect and quadruple-effect absorption refrigeration cycles are still under development and will provide COPs up to 2.

There are other options available in order to increase absorption system's performance such as generator/absorber heat exchanger (GAX) absorption refrigeration cycle and absorption refrigeration cycle with absorber heat recovery. Srikhirin et al. also provides more detail information on the other options to increase system performance [15].

2.2.2. Desiccant refrigeration

Desiccant cooling system is an extension of the evaporative cooling concept in which heat is absorbed by liquid from the substances in contact with it, during evaporation process. There is also a close relation with the latent heat of vaporization of that liquid and heat absorbed during evaporation. The difference between wet bulb and dry bulb temperatures increases the potential of evaporative cooling.

Daou et al. defined desiccants as "natural or synthetic substances capable of absorbing or adsorbing water vapour due the difference of water vapor pressure between the surrounding air and the desiccant surface" [17].

Thus, desiccants are used to dehumidify the inlet air to obtain dry air which is then cooled and humidified by evaporative cooling. It could be followed by a vapour compression or any other cooling system if a sensible cooling is required. The vapour compression system's energy demand decreases as desiccant cooling is based on the consumption of handle latent loads. And also energy savings may reach up to 80% for dry climates since the desiccant cooling system's performance is strongly linked to weather conditions [17].

Fig. 5 shows a schematic of a desiccant cooling system. The air stream from atmosphere is passed through the desiccant wheel at state 1 and significantly dehumidified at state 2. The temperature of air stream increases since adsorption or absorption of water vapour is an exothermic reaction and should be decreased by heat wheel. Heat wheel can be used in conjunction with vapour compression system if required cooling is not supplied. At the end of this state, the temperature of the air stream is more excessively decreased and evaporative cooling is applied according to thermal comfort conditions to increase the humidity ratio of the air. Between state 7 and 8, regeneration of the desiccant material carried out required high temperatures by using a heater. The exhaust air stream can be heated up with solar energy or waste heat.

The COP of the desiccant cooling system is defined by speculation on no demand for a vapour compression system,

$$COP = \frac{Q_L}{Q_H} \tag{2.3}$$

where Q_L is the heat extracted from the conditioned space and Q_H is the heat added into the heater.

The system presented in Fig. 5 is employing solid desiccants while liquid desiccants are also available in other systems [17].

2.2.3. Ejector refrigeration

Another alternative to the conventional vapour compression cycle is ejector refrigeration system which is a thermally driven technology by low grade thermal energy. They have been developed by replacing the compressor with a boiler, an ejector and a pump in the vapour compression cycle [18]. They have a much lower COP usually under 0.3 than vapour compression systems; however offer advantages of simplicity, no moving parts, low operating and installation cost. Their utmost advantage is capability of producing refrigeration from waste heat or solar energy by using heat source at temperatures above 80 °C [19]. A schematic illustration of the system is offered in Fig. 6.

COP of the ejector refrigeration system is defined as:

$$COP = \frac{Q_L}{Q_H} \tag{2.4}$$

where Q_L is the heat extracted from the evaporator and Q_H is the heat added into the boiler.

2.2.4. Rankine refrigeration

The Rankine refrigeration utilizes the power that is generated in the turbine according to the Rankine cycle to drive the compressor in the vapour compression cycle by combining the

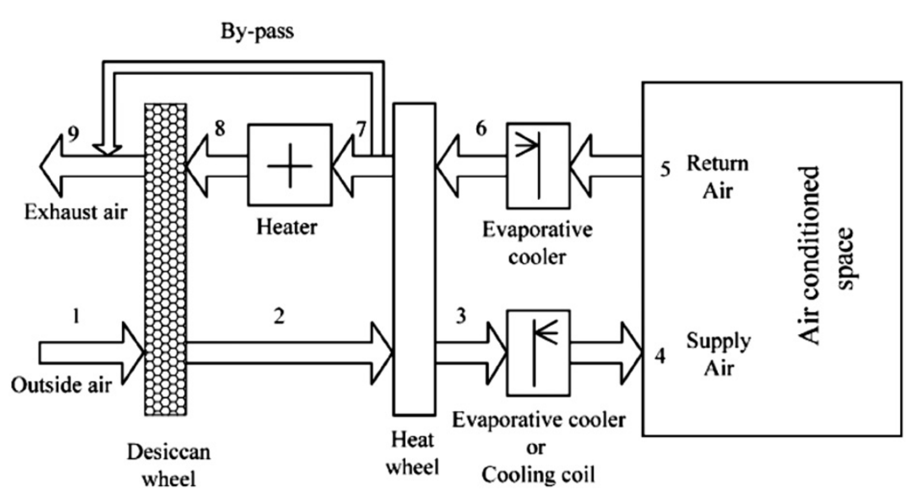

Fig. 5. Desiccant cooling system [17].

Rankine power cycle and vapour compression refrigeration cycle system. A Rankine refrigeration cycle schematic is illustrated in Fig. 7.

The COP of Rankine refrigeration system is the same with vapour compression cycle, whereas the efficiency of Rankine power cycle is directly related to the temperatures of the sink and source. The overall system performance can be amplified by using high efficiency flat plate collectors, evacuated tube collectors or parabolic trough collector. These systems are substantially suitable for large air conditioning applications because of their complexity [18].

2.2.5. Adsorption refrigeration

The interest in adsorption systems has started with energy shortage and ecological problems firstly, and drawing the attention to it since the release of refrigerants such as CFC, HCFC into the atmosphere deplete the ozone layer. Michael Faraday developed the first adsorption cooling system in 1848, which employed ammonia and silver chloride (AgCl) as the working pair. Even the commercial products were made for sorption systems; the invention of cheap and reliable compressors made them unfeasible [21]. Since adsorption refrigeration is similar to absorption refrigeration, the difference between absorption and adsorption refrigeration systems should be emphasized. Basic adsorption refrigeration systems do not need for a pump and a rectifier. The pressure difference arising inside the system is the result of the transfer of a substance from one phase (i.e., vapor) followed by condensation on the surface. They are ideal for industrial air conditioning, process cooling, and waste heat recovery applications and in commercial use as well. During the adsorption process the electricity consumption is minimal.

The absorption refrigeration system uses a solution containing water and lithium bromide salt to absorb heat from the surroundings and driven by hot water, steam, or combustion. The substance is transferred from one phase to another and penetrates the second substance to form a solution. The most common sources of commercial climate control and industrial machinery cooling systems working principle are reliant on the absorption refrigeration system. The cooling process is environment friendly since no use hazardous gases that deplete the ozone layer such CFC's or ammonia.

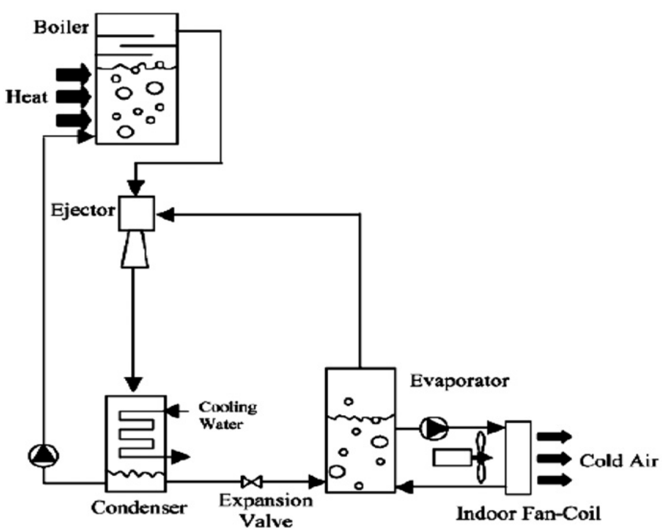

Fig. 6. Ejector refrigeration system [18].

3. Thermodynamic analysis of absorption cooling and desiccant cooling system

The technologies which allow utilizing of solar thermal collectors for air-conditioning systems can be divided into two main categories; closed and open cycles systems.

- Closed cycle (absorption and adsorption refrigeration) is employed to produce chilled water that might be used for various air-conditioning equipments.
- Open cycle (desiccant cooling system) is utilized for direct treatment of air in a ventilation system [22].

3.1. Closed cycle systems

These types of systems are commonly based on absorption cycle (absorption air conditioning system) which basically comprises an evaporator, a condenser, an absorber and economizer.

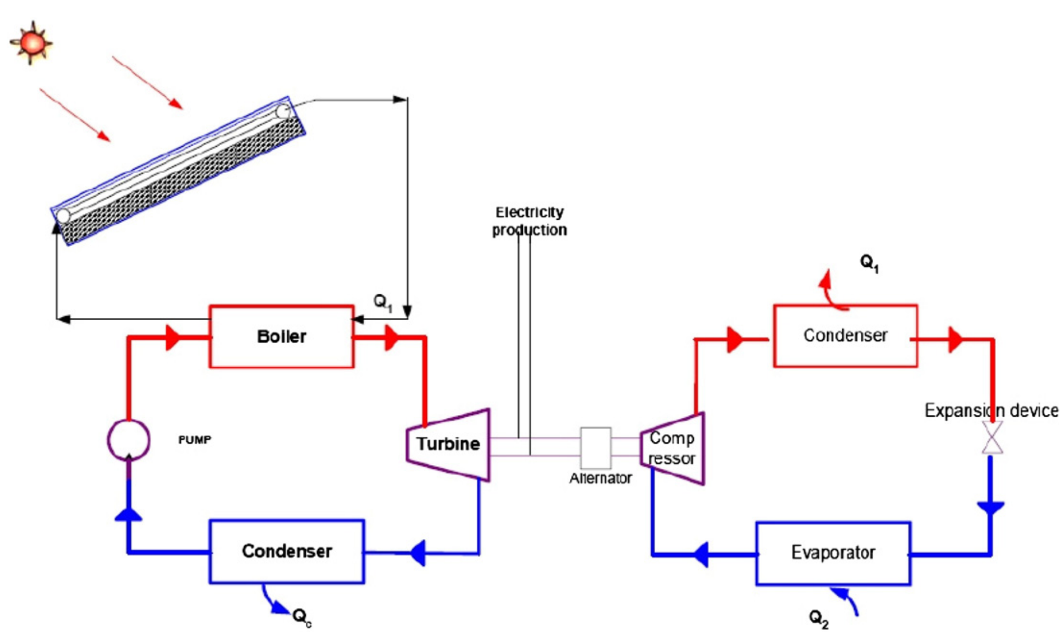

Fig. 7. Rankine refrigeration system [20].

Briefly, single effect absorption system uses a refrigerant spreading from a condenser to an evaporator, as similar to the method in the conventional compression but it differs in the pressurization stages, [23].

In general, at low pressure, evaporation of water extracts heat from the environment (usable cold produced) and then the compression of the water vapour desorbs in the condenser at high pressure, when heat is supplied to the desorber. As it can be seen in Fig. 8, the system operates with two pressure levels and three temperature levels [24]:

- High temperature—heat supply in the desorber.
- Intermediate temperature—heat rejection in the absorber.
- Low temperature—cooling in the evaporator.

There are two most common combination of fluids that include lithium bromide–water ($LiBr–H_2O$) and ammonia–water ($H_2O–NH_3$). The thermal coefficient of performance (COP), which defines the performance of thermally driven chillers, is described as the production cold per unit driving heat (Fig. 9, Eq. (3.1)), [25]. This rate for single-effect absorption system is limited in COP approximately 0.7 for $LiBr–H_2O$ and around at 0.6 for $H_2O–NH_3$ [7]. It is reported that $LiBr–H_2O$ has higher COP than the other working fluids. Although, it has some disadvantages such as limitation range of operation, its low cost and great performance makes it favourable absorbent-refrigerant pair for use in solar cooling systems. For this reason, $LiBr–H_2O$ is preferred to use for most solar-absorption air conditioning applications [1,26].

$$COP_c = \frac{\text{Desired output}}{\text{Required input}} = \frac{Q_L}{W_{net,in}} \quad (3.1)$$

where $W_{net,in}$ is the net work input to the chiller.

$$W_{net,in} = Q_H - Q_L \text{ [kJ]}$$

Q_L is the magnitude of the heat removed from the refrigerated space at temperature T_L; Q_H is the magnitude of the heat rejected to the warm environment at temperature T_H.

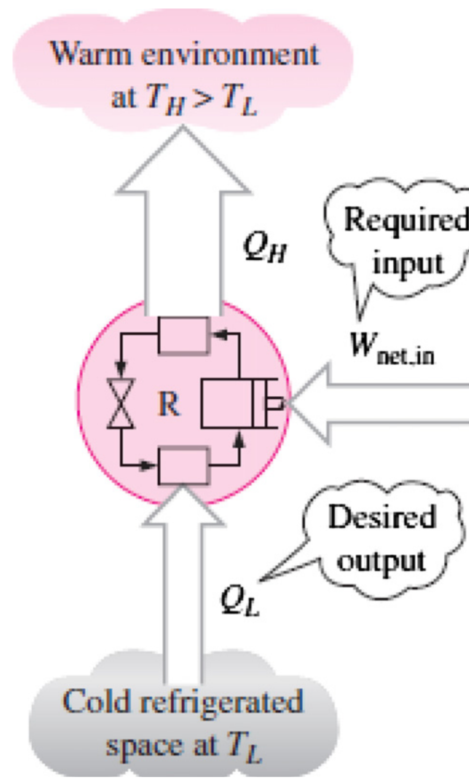

Fig. 9. Basic thermodynamic scheme of a chiller is to remove Q_L from the cooled space [25].

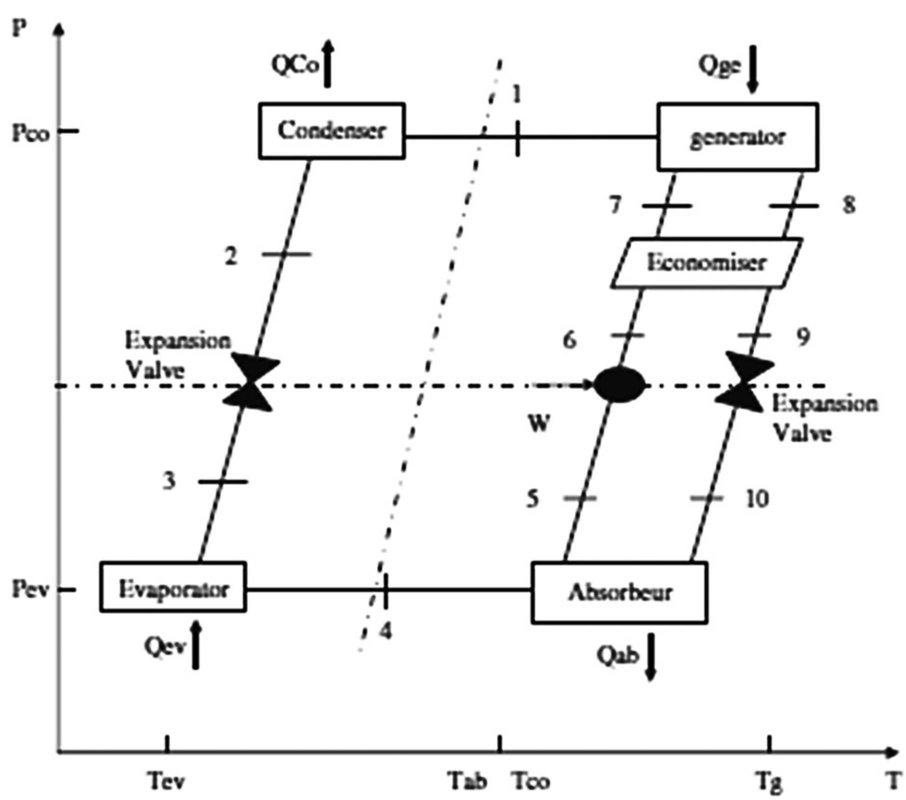

Fig. 8. Pressure against temperature of an absorption cooling system [24].

However, these systems require large solar collector area to capture much heat needed for their operation. Once COP could be improved, which might be achieved using a higher temperature heat source, this collector area can be reduced [7,27].

While single effect systems have been used on most solar-powered absorption cooling projects with low temperature, presently, double effect systems, with development of gas-fired absorption system particularly in the USA and Japan, are available on the market with COP in the range 1.0–1.2. Furthermore, triple effect systems with COP of about 1.7 are still under development but close to market now. Fig. 10 shows the COP_{ideal} (carnot) with COP-value of multi effect thermally driven chillers with the same component size and under the same operation conditions.

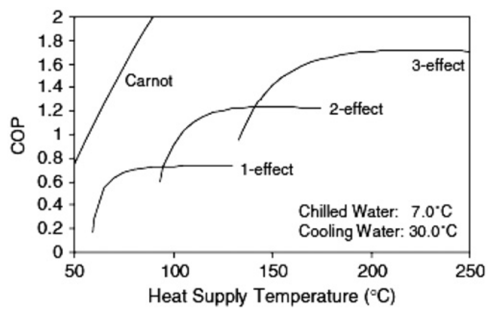

Fig. 10. COP as a function of (solar) heat supply temperature for single-double- and triple effect LiBr–H_2O absorption chillers, [5].

As it is shown single-effect system offers best results in the temperature between 80 and 100 °C. Although large capacities absorption chillers (several thousand watts) are available on the market from various manufacturers, only few small capacities systems (< 100 kW) are available in the market but are expected to increase the quantity of small scale systems in the market. It is known that further research and improvement for these small absorption machines is necessary to reduce their volume and increase the power density.

3.2. Open-cycle systems

Desiccant cooling systems (DCS), which are especially suitable for solar thermal application due to the low temperature demands around 60–80 °C, are a fixed technology for air-conditioning buildings. Unlike thermally driven chillers, open cooling cycle sorption technology does not produce chilled water. This technology produces directly conditioned air, which is based on the air dehumidification by an absorbent as lithium chloride or silica gel. Today, the standard open cycle uses rotating sorption

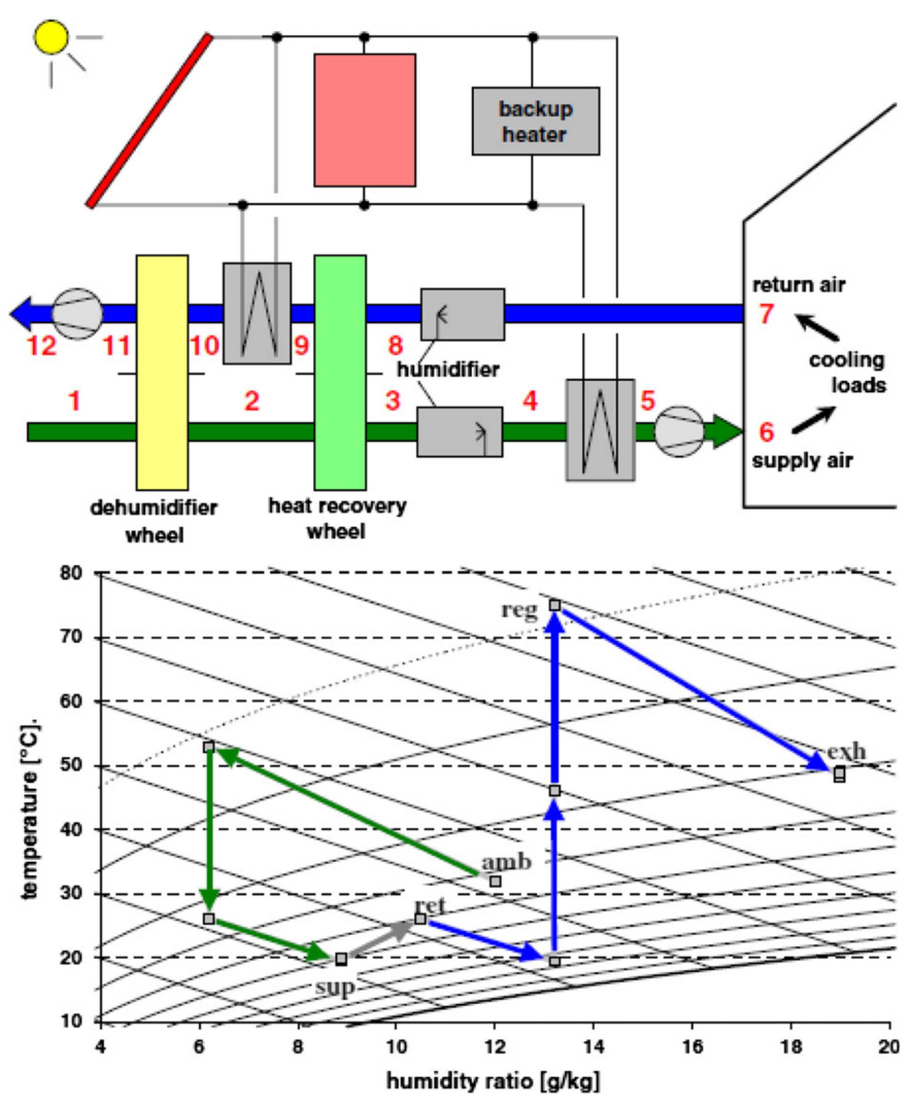

Fig. 11. Process steps and general structure of open cycle desiccant system and T–x diagram of humid air (bottom), [22].

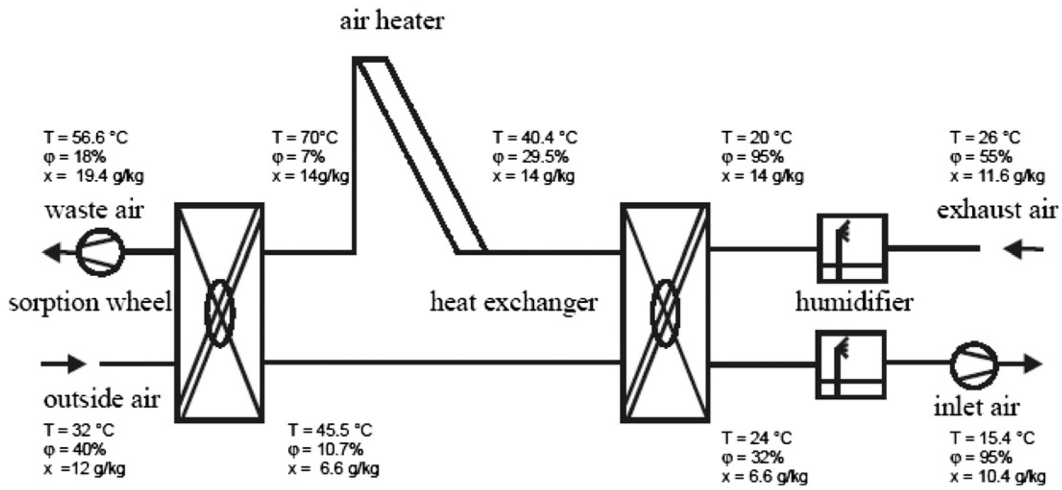

Fig. 12. Humidity and temperature conditions in a open cycle system, [28].

wheels, where the outside air humidity captured in the absorbent and then carried out to the exit air heated by waste heat. Standard desiccant cooling system is shown on Fig. 11.

The cycle process that the air follows is explained in the following sections;

1-2 Dehumidifications from the outside air by an absorbent:
Adiabatic process.
Outside air is dried in the sorption wheel.
2-3 Pre-cooling of the air in the heat recovery device.
3-4 Evaporative cooling produces the desired supply air status.
4-5 Pre-heating of air.
5-6 Fan cause to increase temperature slowly.
6-7 Supply air temperature and humidity are increased.
7-8 Evaporative cooling of return air from the building.
8-9 Pre-heated for return air.
9-10 Solar thermal collector provide regeneration heat.
10-11 The water bound of the dehumidifier wheel is desorbed.
11-12 Exhaust air is released to the environment (return air fan).

Typical humidity conditions and temperature at design conditions of 40% and 32 °C are illustrated in Fig. 12.

4. Examples of small scale solar cooling installations

4.1. Solar Info Center (SIC) in Freiburg, Germany (10 kW, 2004)

A part of the solar info center building, seminar room and area of offices, is air-conditioned by a liquid desiccant cooling system. This system was a pilot installation to demonstrate the applicability of this technology. The technology used solar thermal heat, from flat plate solar collectors, for regeneration of the diluted LiCl solution. The sorption section is cooled that leads to increase the sorption efficiency. A positive effect of this system is that it allows a decoupling in time between the regeneration and dehumidification process to a certain degree, [29,30].

System performance and improvements, overall cold production from whole operation modes, desiccant cooling, adiabatic cooling and free cooling, demonstrate the average COP as 1.0 and annual collector yield as 270 kW h/m^{2}a were achieved. During the operation, several improvements in the system were implemented such as some optimizations of fans and pumps. Tables 1–3 and Fig. 13 show more details for the system, [30].

Table 1
Central air-conditioning unit.

Technology	Open cycle (DEC)
Nominal air volume flow rate	1500 m^3/h
Minimum air volume flow rate	600 m^3/h
Desiccant cooling system type	Liquid desiccant
Desiccant type	Lithium chloride
Cooling capacity	10 kW
Brand of desiccant unit	Menerga

Table 2
Solar thermal.

Collector type	Flat-plate
Brand of collector	Ufe Ecostar
Collector area	16.8 m^2 aperture
Tilt angle, orientation	30°S
Collector fluid	Water
Typical operation temperature	55–70 °C regeneration temperature

Table 3
Configuration.

Heat storage	1.5 m^3
Cold storage	Solution storages
Auxiliary heating support	District heating network
Use of auxiliary heating system	Supply air heating in winter
Auxiliary chiller	No

4.2. Fraunhofer ISE, Freiburg, Germany (5.5 kW, 2007)

The system has been installed as a small scale thermally driven chiller application by Institute for Solar Energy System (ISE). Briefly, the technology is a closed cycle chilled water system for cooling canteen kitchen at Fraunhofer ISE. Solar thermal system and heat network of the Institute provide driving heat. The technology can be operated in three modes; cooling mode in summer, heating of the chiller is rejected in intermediate temperature by three ground tubes, and also ground tubes are used as a low temperature in winter (see Fig. 14), [31].

When the system performance was examined, from August 2008 to July 2009, value of COP was noted as an average 0.43. As an advantage for the system, cooling tower is not required and

Fig. 13. Solar Info Center building. Type of building seminar room and office area within the SIC location Freiburg, Germany (48°N, 7°50′E) in operation since 2004 system operated by Fraunhofer ISE air-conditioned area 300 m^2.

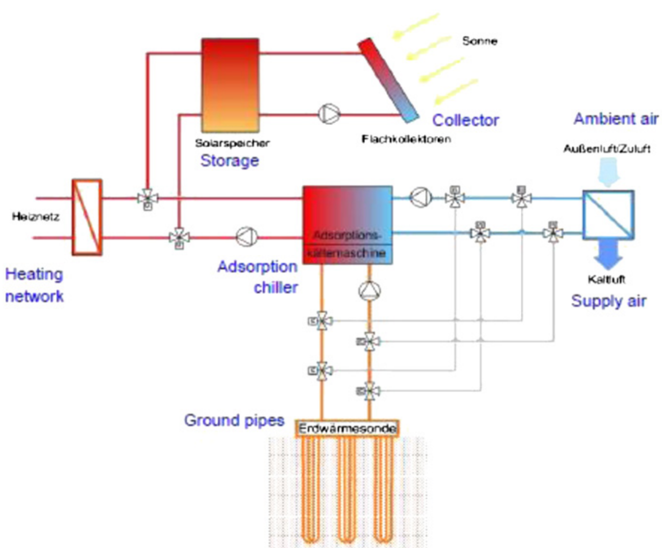

Fig. 14. Scheme of the system during the summer operation mode.

Table 4
Central air-conditioning unit.

Technology	Closed cycle
Nominal capacity	5.5 kW_{cold}
Type of closed system	Adsorption
Brand of chiller unit	SorTech ACS 05
Chilled water application	Supply air cooling
Dehumidification	Occasionally
Heat rejection system	Dry, ground tubes

that leads to cost reduction. Also, the technology is the noiseless operation of the chiller. More details are given in Tables 4–6 and Fig. 15, [32].

4.3. *Residence du Lac, Maclas, France (10 kW, 2007)*

In this project, only small part of the Residance du Lac building, leisure space/restaurant area that is about 210 m^2, was proposed to cool with a closed cycle sytem by Syndicat Intercommunal d'Energie de la Loire (SIEL). Mainly, the system was designed using evacuated tube collectors with an absorption chiller of 10 kW. The system can operate on both cooling (June to mid September) and heating (mid October to end of May) modes. The heat rejection system was done by drycooler located in the Northern of the building. The system performance values have not been monitored yet. However, energy production estimation for cooling mode is about 4300 kW h/year (4 months) and also energy saving for cooling is expected to be around 5 c€/kW h (EER=2; bad quality split). Additionally, total electricity consumption is expected to be around 845 kW h=42 €/year. More details and the system values can be seen in Tables 7–9 and on Fig. 16, [33].

Table 5
Solar thermal.

Collector type	Flat-plate
Brand of collector	Solvis FF 35s 3/2 FKY
Collector area	22 m^2 aperture
Tilt angle, orientation	30°S
Collector fluid	Water–glycol
Typical operation temperature	75 °C driving temperature for chiller operation

Table 6
Configuration.

Heat storage	2 m^3 water
Cold storage	None
Auxiliary heating support	Institute heat network, operated by CHP and gas boiler
Use of auxiliary heating system	Auxiliary driving source for chiller, auxiliary driving source for heat pump operation in winter
Auxiliary chiller	No

Fig. 15. Fraunhofer ISE institute building. Type of building kitchen area of institute; location Freiburg, Germany; in operation since 2007; system operated by Fraunhofer ISE; air-conditioned area 42 m^2.

Table 7
Central air-conditioning unit.

Technology	Closed cycle
Nominal capacity	10 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	Sonnenklima
Chilled water application	Fan coils
Dehumidification	No
Heat rejection system	Dry

Table 8
Solar thermal.

Collector type	Evacuated tube
Brand of collector	Thermomax Mazdon 20
Collector area	24 m^2 absorber area
Tilt angle, orientation	30°, 15°W
Collector fluid	Water–glycol
Typical operation temperature	75 °C driving temperature for chiller operation

Table 9
Configuration.

Heat storage	0.5 m^3 water
Cold storage	Buffer water (80 L)
Auxiliary heating support	None
Use of auxiliary heating system	–
Auxiliary chiller	Yes
– type	el. compression chiller
– capacity	3 kW_{cold}

Fig. 16. Cooled area of residence du Lac. Type of building retired people residence; location Maclas, France; auxiliary heating support fuel (central heating system); in operation since 2007 system operated by SIEL; air-conditioned area 210 m^2.

Table 10
Central air-conditioning unit.

Technology	Closed cycle
Nominal capacity	7.5 kW_{cold}
Type of closed system	Adsorption
Brand of chiller unit	SORTECH
Chilled water application	Fan coils
Dehumidification	No
Heat rejection system	Dry cooling tower with adiabatic spaying

4.4. CNRS Promes Research Center Office, Perpignan, France (7.5 kW, 2008)

The CNRS Promes Research Center office, where the system is installed, is located on the border of TECNOSUD technical area in Langyedoc Roussillon (south of France). The solar cooling system, which is fed by 24 m^2 double glazed flat plate collectors that carry out thermal energy used for driving a 7.5 kW adsorption chiller, is installed on the ground floor for a small part of the building.

Table 11
Solar thermal.

Collector type	Double glazed flat plat collectors
Brand of collector	Schüco
Collector area	25 m^2 absorber area
Tilt angle, orientation	30°, 45°E
Collector fluid	Water
Typical operation temperature	75 °C driving temperature for chiller operation

Table 12
Configuration.

Heat storage	0.3 m^3 water
Cold storage	0.0033 m^3 water
Auxiliary heating support	None
Use of auxiliary heating system	None
Auxiliary chiller	Yes
– type	el. compression chiller

Fig. 17. Collector field for the system. Type of building office building; Location Perpignan; in operation since July 2008; system operated by Neotec; air-conditioned area 180 m^2; capacity 7.5 kW.

The plant is generating energy through multi split compression chiller system. Also, heat rejection system, is used only on very hot days, and is done by drycooler (spring water spraying device). More details are revealed in Tables 10–12 and on Fig. 17, [34].

According to the results of the plat after one year (2008–2009), energy production was monitored for cooling as 2500 kW h/year (5 months on 12) and energy saving was measured for both cooling and heating as 10 c€/kW h (ESEER=2; average quality multi split). On the other hand total electricity consumption was calculated around 580 kW h=58 €/year.

4.5. University Hospital in Freiburg, Germany (70 kW, 1999)

Another system which is also installed in Freiburg is an adsorption chiller founded on close cycle solar cooling system, and it is applied by the University hospital for air-conditioning of a laboratory. The capacity of the system is about 70 kW and aperture area for collectors around 170 m^2. The project was observed over 4 years. Main results exhibit that tube solar collectors used in the system function well with favourable performance. In addition, COP of the adsorption chiller seems reasonable with additional improvements in control. At the end of the 2003, an average thermal COP was calculated as 0.42 and annual specific yield was monitored as 365 kW h/m^2a. Nonetheless, electricity consumption for this system is significantly high. Operation values of the system are presented in Fig. 18. It can be clearly seen that solar system absorbed most of the driving heat during the daytime hours at clear sky in summer. Fig. 19 shows the collector area for this project and also more details are seen in Tables 13–15, [35].

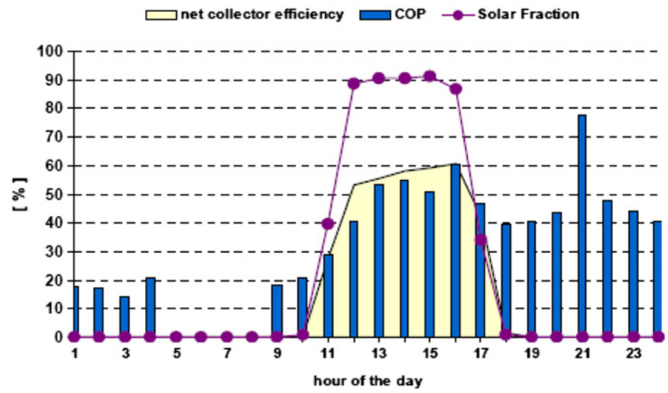

Fig. 18. System performance in a summer day (18 August 2004), [22].

Fig. 19. Solar collector area for the system in University hospital in Freiburg, [22]. Type of building Laboratory building of the University Hospital; location Freiburg, Germany (48°N, 7°50′E); in operation since 1999; system operated by Energy Management of the University Hospital; air-conditioned area 360 m^2.

Table 13
Central air-conditioning unit.

Technology	Closed cycle
Nominal capacity	70 kW$_{cold}$
Type of closed system	Adsorption
Brand of chiller unit	Nishiyodo NAK 20/70
Chilled water application	Supply air cooling
Dehumidification	Occasionally
Heat rejection system	Closed wet cooling tower

Table 14
Solar thermal.

Collector type	Vacuum tubes
Brand of collector	Seido 2-16
Collector area	167 m^2 aperture
Tilt angle, orientation	30° and 45°S
Collector fluid	Water–glycol
Typical operationtemperature	75 °C driving temperature for chiller operation

In addition to these mentioned applications above, many installations are still in use today. Some of them are outlined in Table 16.

Table 15
Configuration.

Heat storage	6 m^3 water
Cold storage	2 m^2 water
Auxiliary heating support	Condensating steam heat exchanger, driven by the hospital steam network
Use of auxiliary heating system	Auxiliary driving source for chiller, auxiliary driving source for supply air heating in winter
Auxiliary chiller	No

5. Recent developments of solar thermal cooling technology

5.1. New chiller technology

Currently, some studies on solar assisted air conditioning systems have been applied to provide small scale solar thermal cooling applications. One of them is a chiller based on the steam jet cycle which is modified into small size units (20–200 kW cooling power) to be combined with solar thermal technologies. When compared to the absorption chillers, COP values for this system may exceed the value 1.0. The system does not have moving parts, therefore construction principle is simple. Moreover, only water is used in the fluid cycle. The driving temperature for steam jet cycle is offered around 200 °C. Hence, concentrating solar collectors with tracking systems are required to capture much more sun lights. However, in present studies the technology may be economically competitive in comparison with the other cooling systems. Further explorations and development work on cost reduction is necessary in order to provide series of fabrication of adapted steam jet systems, [46].

5.2. Improvements on solar thermal cooling equipment

In recent times, several developments have occurred to improve new thermally driven cooling equipment. Here, some of them are elucidated;

- Many research and development (R&D) activities focus on the improvement of low cooling capacities (< 50 kW down to less than 5 kW) for thermally driven water chillers. For instance, different types of liquid sorption pairs such as lithium-bromide/water or ammonia/water as well as solids such as zeolite/water and silica gel/water. Recent developments are summarized in Table 17.
- Several R&D studies focus on the development of liquid desiccant of open-cycle that may operate with low rate solar heat. When liquid desiccant is used, cooling sorption process is operated easily with higher dehumidification and low regeneration temperature compared to standard DCS.

Another advantage of using liquid desiccant is that the system provides separate absorption and regeneration processes at the same time. Thus, absorbent–refrigerant pair can be used as chemical storage; 'in order to success high storage densities is to install high efficient absorption system that cause of large differences between concentration of concentrated and diluted solution, [22].

- According to another R&D projects, solid sorption material can be used to capture a large dehumidification of the process of air.

5.3. New development of absorption heat pump for cooling system

SorTech AG and Fraunhofer ISE project is a small prototype absorption heat pump based on fluid pair silica gel/water. It is possible that project will be installed in the near future because

Table 16
Recent installations with their main features.

Plant	Operation since	Technology	Air-conditioned/ collector area (m^2)	Capacity (kW)	System operator
Manufacturing area in Bolzano, Italy [36]	2005	Closed cycle, adsorption system	400/150	15	EURAC
Residential building, Milan [37]	2007	Closed cycle, adsorption system	90/25 (aperture)	4.5	Politecnico di Milano
Absorption chiller solution, Sattledt, Austria [38]	2005	Closed cycle, adsorption system, chilled ceiling	350/40 (aperture)	15	SOLution Solartechnik GmbH
Residential solar cooling and heating, Derio, Spain [39]	2007	Closed cycle, adsorption system, radiant floor	200/21.6 (aperture)	10	Lansolar
Residential building in Thening, Austria [40]	2007	Closed cycle, adsorption system, chilled wall	177/40 (gross)	5.5	Building owner
Office building of S.O.L.I.D., Graz, Austria [41]	2008	Closed cycle, adsorption system, cooled ceilings elements	485/60 (gross)	17.5	SOLID
Office building of IBA AG in Fürth, Germany [42]	2007	Closed cycle, adsorption system, chilled ceilings, fan coils	920/87.7 (aperture)	30	IBA AG
Office building Vajra in Loulé, Portugal [43]	2005	Closed cycle, adsorption system, fan coils	670/128.8 (aperture)	35	Vajra
Technical College for Engineering in Butzbach, Germany [44]	2008	Closed cycle, adsorption system, supply air cooling and dehumidification, chilled ceilings	335/60 (aperture)	20	Technical college
Town Hall and Service Center in Gleisdorf, Austria [45]	2008	Open cycle (DEC), solid desiccant, Silica Gel and closed cycle, adsorption (water/lithium bromide)	2000/302 (gross)	35	Community of Gleisdorf

Table 17
Recent developments for thermally driven cooling equipment.

	Working fluid	Sorption material	Driving temperature (°C)	Key features	Developers	Reference
Closed cycles	**Water**	**Lithium-bromide**	70–95	Rotating absorber; very low temp. on HXs	Rotartica research center	Ikerlan [47]
Closed cycles	**Water**		80–90	Available in market (> 15 kW)	Company EAW	Safarik [48]
Closed cycles	**Water**		70–95	Compact design, prototype	Company Phönix	Kühn et al. [49]
Closed cycles	**Water**		75–95	Directly air cooled, in research	Polynechnic Uni.	Task 25 [50]
Closed cycles	**Water**	**Silica gel**	65–95	Compact, no moving part	Company Sortech	Nünez et al. [51]
Closed cycles	**Water**	**Lithium-chloride**	70–100	High efficient storage	Company climate well	Baled et al. [52]
Closed cycles	**Water**	**Sodium-sulfide**	80–90	High efficient storage	Company sweat	De Boer [53]
Closed cycles	**Ammonia**	**Water**	100–120	Dry air cooling	Company Aosol	Afonso et al. [54]
Closed cycles	**Ammonia**		80–110	Institue Joenneum	Adjustable various type	Podesser [55]
Closed cycles	**Ammonia**		70–120	No solution pump	University of Stutgart	Task 25 [56]
Open cycles	**Water**	**Lithium-chloride**	60–90	Liquid sorption-indirect evaporative cooling	Company Menergy	Röben [57]
Open cycles	**Water**	**Lithium-chloride**	60–90	Liquid desiccant	Technion Haifa	Gommed et al. [58]
Open cycles	**Water**	**Lithium-chloride**	60–90	Liquid desiccant	Research Center ZAE	Lavemann [59]
Open cycles	**Water**	**Silica gel**	60–100	High efficient cooled and dehumidification	Fraunhofer Institue	Motta et al. [60]

EU supports this project MODESTORE (Modular High Energy Density Sorption Storage). Fig. 20 shows structure of the system.

On this system, when temperature of buffer storage is high enough for driving the absorption system, cooling can be provided.

Thus, evaporator is settled on the cooling surface of the building. Moreover, ground couple heat exchanger provides direct cooling and rejects the heat from ground. A ground coupled heat exchanger is employed as heat sink especially in summer.

5.4. Sorption dehumidifier for high efficiency

Using rotating sorption wheels in the DCS has several drawbacks;

- Leakage between supply air and regeneration air leads to reduce performances of rotor technology, particularly at processing small capacity systems.
- Adiabatic process, which is not cooled, causes a reduction in the dehumidification capability of desiccant material.
- In the standard desiccant cooling system, indirect evaporative cooling is employed. However, this system cannot capture all high potential of enthalpy of the building regeneration air.

Indirect evaporative cooled sorptive heat exchanger (ECOS) is a new desiccant concept which aims to solve the previously mentioned issues. The design of the process outputs shows that dehumidification for this system is far higher than the conventional systems. This project is planned to be established in the high ambient air humidity places such as Mediterranean and tropical regions. Also, the system structure is not complex because there is no rotating part that is necessary in standard system.

Mainly, the system consists of two simultaneous processes; sorptive dehumidification and indirect evaporative cooling of the supply air. Furthermore, in order to continue humidification process and obtain high heat exchange potential, indirect evaporative cooling is used on regeneration air side of the heat exchanger. In this system, counter-flow (air-to-air) heat exchanger technology has been applied (see Fig. 21). The heat exchanger is separated as cooling (grey line in Fig. 21 top) and sorptive (black line) part but both of them are in thermal contact. Supply air is dehumidified in the sorptive side and humidification of the cooling stream continues in the cooling channel.

All the system comprise of two sorptive heat exchangers, managed regularly. Simply, while one component is employed in the air-conditioning process, the other one is regenerated and

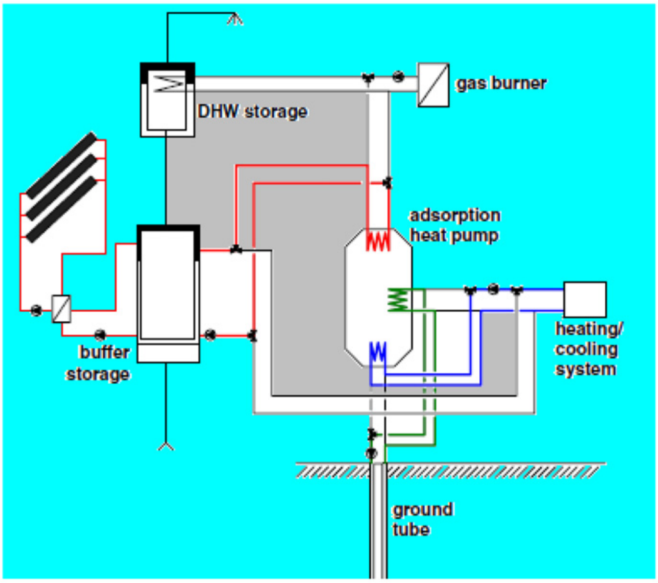

Fig. 20. Absorption heat pump system for solar cooling, [22].

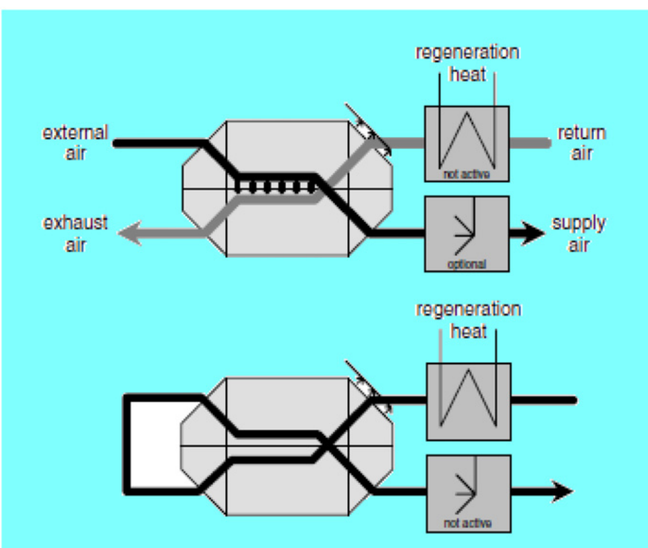

Fig. 21. Evaporative cooled sorptive heat exchanger system during air humidification (top) and regeneration (bottom), [22].

pre-cooled before the following air-conditioning process. According to the Motta, Henning and Kallwellis' experiments; the temperature of the process air is reduced from about 32–36 °C at the inlet to about 21 °C at the outlet, [59].

6. Subsystems of solar thermal cooling technology

This chapter provides a detailed view of an integrated model of the solar absorption cooling system. Whereas a comprehensive knowledge is available in planning and designing solar cooling systems, there is a lack of knowledge and standard of whole system planning for solar cooling systems. Recently published some useful books are utilized as reference in integrated modeling of the system [61,62].

6.1. Solar-thermal cooling system concepts

Regarding the solar energy, fraction is the starting point for solar cooling system [61]. Solar energy fraction that will be named shortly solar fraction (SF) hereafter is the ratio of solar energy used in the whole system divided by the total energy requirement of the solar cooling plant. There are two types of system design in principal:

- Solar-thermally autonomous systems.
- Solar-assisted systems.

Solar-thermally autonomous systems are self-sufficient systems where the sun is used for extracting the all necessary heat for thermally driven cooling system. Since we do not have the ability to control the sun and weather conditions, the desired indoor conditions may not be continuously satisfying with increasing comfort expectations and increasing cooling loads in this type of systems. Thus, indoor temperature and humidity are statistically analyzed to see how often they exceed confident comfort necessities. The synchronization of this system should be very useful in terms of the cooling loads and solar gains. Solar-thermally autonomous systems should be conceived when a back-up system is not feasible.

Solar-assisted systems offer opportunities to reduce the increasing conventional energy usage for the air-conditioning demand in buildings in an energy-efficient way by using solar energy. The potential of this technology is realized firstly for solar collectors for domestic hot water systems which is now far from point of released. These systems also make use of back-up systems to supply the required amount of cooling for development of solar-assisted air-conditioning systems. Every individual project has its own criteria in the selection of the back-up system. Either a second heat source such as a boiler or heat-driven chillers could be a back-up system for direct generation of cooling power.

Among the solar heat driven chiller based systems, the most widely used system is this configuration [61].

6.2. Subsystems of the solar-thermal cooling system

During the last decade more companies as system providers for solar-thermal cooling in the solar business have taken place on the market. These companies offer solar cooling systems with four different major subsystems [62] as:

- Building (thermal insulation).
- Air-conditioning system (fan-coils, air handling unit, chilled ceilings, etc.)
- Cold-supply circuit (chiller, cold storage tank, etc.)
- Heat-supply circuit (solar collectors, heat storage tank, auxiliary heater, etc.)

The use of these subsystems depends on the requirements and climatic zone, and some of them may not be feasible. For example, an active air-conditioning of buildings requires humidification/dehumidification that is almost necessary at climate conditions and detailed design of the building. Thus, the scope of this study investigated the other three subsystems, since the air-conditioning system involves many detailed steps.

6.2.1. Building

The dominating energy consuming services in buildings are mainly high energy demand systems such as heating, cooling and lighting load, therefore an increasing demand on the design of systems that incorporating energy efficiency, renewable and

sustainable green energy and requires interdisciplinary cooperation between architects and engineers.

Energy-efficient building designs reduce the pollution generated by energy production and also reduce wasting of the resources. Hence, whole-building approach does not have the priority relation in construction cost, and purpose of the building is neither to save nor use energy.

Developed countries have standardized the progressive design strategies and building energy performance, and updating the energy efficiency requirements continuously. ISO 13790[1] and ASHRAE 90.2-2007[2] are the most widely used ones related to these regulations.

Zero net energy consumption and zero carbon emissions principle is becoming popular in the developed countries and they claim to design buildings via renewable energy harvesting by 2050. The reason is that cutting of greenhouse gas emissions are directly related to energy consumption of buildings that consume 40% of the total fossil energy in USA and Europe [63].

6.2.2. Heat supply circuit

The heat production sub-system consists of solar collector, which is main the component and thermal storage unit. These components, especially collectors are mentioned in the following lines, which provide heat to a thermally driven cooling system.

6.2.2.1. Solar collector and back-up heat sources. The solar collector is the main part of the sub-system to convert solar energy to the thermal energy that conducts the solar-assisted cooling system. In this part, different types of solar collectors and their features are presented and discussed.

There is an absorber, central component, in each solar collector. Solar radiation is converted into heat; certain amount of this heat is released to the environment and the others are turned into fluid for heat transfer.

Energy balance of a solar thermal is demonstrated in the following equation:

$$A \times G = \dot{Q}_{use} + \dot{Q}_{loss,opt} + \dot{Q}_{loss,convective} + \dot{Q}_{loss,conductive} + \dot{Q}_{loss,radiative} \tag{6.1}$$

where A is the absorber area (m^2); G is the incident global solar radiation on the collector surface (W/m^2); $\dot{Q}_{use}$ is the useful heating power of the collector (W); $\dot{Q}_{loss,opt}$ denotes the optical losses (W); $\dot{Q}_{loss,convective}$ expresses connective losses (W); $\dot{Q}_{loss,conductive}$ expresses heat conduction losses (W); $\dot{Q}_{loss,radiative}$ denotes radioactive losses (W).

6.2.2.1.1. Collector efficiency. The solar collector efficiency is known as ratio of $\dot{Q}_{use}$ to the global radiation incident (G) on the collector area (A):

$$\eta = \frac{\dot{Q}_{use}}{A \times G} \tag{6.2}$$

The measurement of a typical solar collector performance is shown with different collector losses (see Fig. 22).

$$\eta = k(\Theta) \times c_0 - c_1 \times x - c_2 \times x^2 \times G \quad \text{where } x = \frac{T_{av} - T_{amb}}{G} \tag{6.3}$$

where T_{av} is the average fluid temperature in the collector; T_{amb} is the ambient temperature; c_0 denotes the optical efficiency; c_1 is the loss coefficient; c_2 is the quadratic loss coefficient.

Additionally, the solar collector efficiency curve is presented (see Fig. 23) for different incident solar radiation values as a function of temperature difference between fluid and ambient ($T_{av} - T_{amb}$).

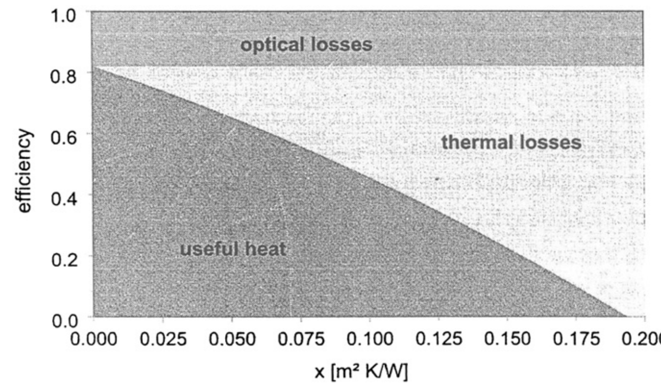

Fig. 22. Typical solar collector efficiency curve [61].

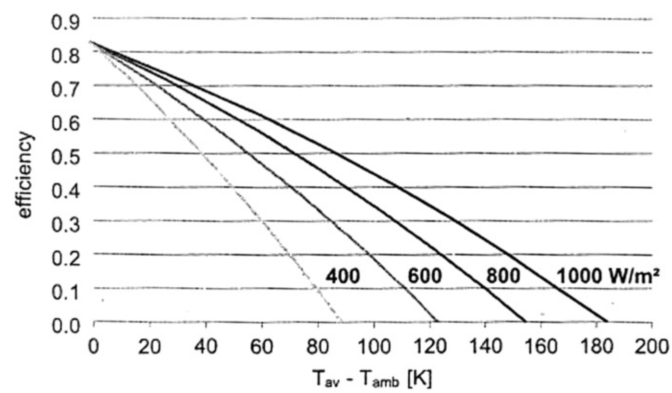

Fig. 23. Common collector efficiency as a function of $T_{av} - T_{amb}$ [61].

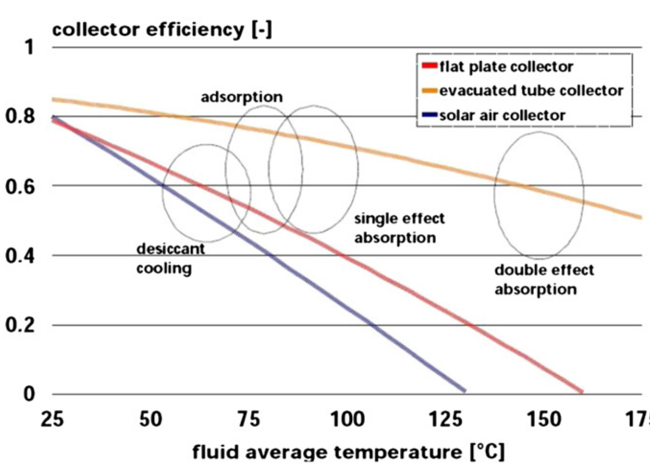

Fig. 24. Operating range for different cooling technologies [61].

Since solar collectors are employed to obtain heat in this study, selection of the collector type is significant to deliver the required regeneration temperature. In addition to this, both the selection of collector type and the area influence performance and economics of the cooling system directly. The main effect for the selection of solar collector type is driving temperature of the chiller. Fig. 24 [61] shows solar collector efficiency for different cooling technology. As it can be seen in the figure, solar collector efficiency should be minimum 50–60% for any solar cooling system.

The driving temperature for the closed cycle solar cooling technologies is in the range of 50 to 90 °C as seen in Fig. 24. As a

[1] Energy performance of buildings—Calculation of energy use for space heating and cooling.

[2] Energy-efficient design of low-rise residential buildings.

result, both evacuated tube and flat plate collectors can be utilized for this temperature range.

6.2.2.1.2. Flat plate collectors. Flat plate collectors dominate the market and they are the most common solar collector type. This kind of collectors can be designed in modules from about 1.8 m^2 up to 10 m^2. The solar collectors are settled up on a simple supporting structure to obtain ideal tilt and orientation. Another advantage of the solar collectors is that they can be integrated into a sloped roof. Typical efficiency curves for different types of flat-plate collectors are plotted in Fig. 25.

6.2.2.1.3. Evacuated tube collectors. An evacuated tube collector comprises single tubes that are connected to a header pipe. In order to reduce heat losses, all single tube is evacuated.

The evacuated tube collectors have three main geometric configurations:

- Concentric fluid inlet and outlet: At this system, each single pipe may be rotated easily, because header pipe is symmetric to rotation that provides obtaining the absorber fin to get the required tilt angle.
- The system has traditional evacuated tube collector structure that has two separate pipes for inlet and outlet.
- The system is called 'Sydney' type collector which is formed double glass tube that includes cylindrical metal absorber. One of the most important advantage for the collectors is that reduce sealing problems because the boundary is all glass.

Typical efficiency curves for evacuated tube collectors are plotted in Fig. 26.

6.2.2.1.4. Discussion. Some differences between flat-plate and evacuated tube collectors are described in following sentences.

For flat-plate collector.

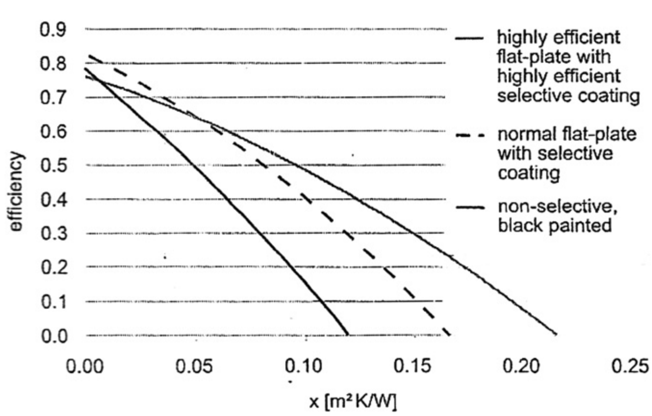

Fig. 25. Typical efficiency curves for different types of flat-plate collectors.

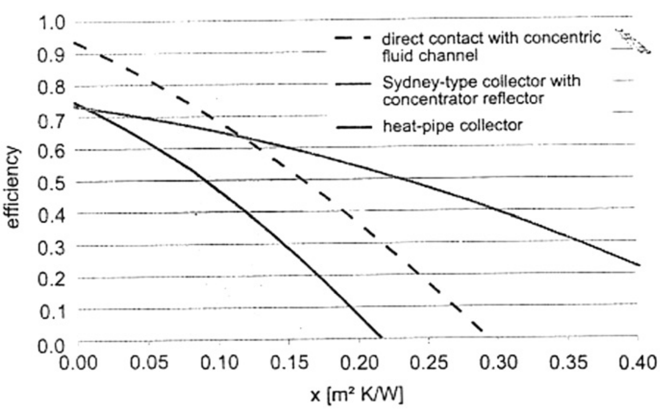

Fig. 26. Efficiency curve for typical evacuated tube collectors [61].

- Especially during the windy and cold days heat losses occur because of the gap between absorber and cover pane. Settled up condensation affects the collector negatively due to the corrosion which causes to reduce performance and durability.
- Installation is difficult. Once one of panels has a lead, the whole collectors have to be shut down and removed.

For evacuated tube collector.

- The collector is isolated from adverse ambient conditions because the collector sealed inside an evacuated glass tube. Thus, convection and conduction does not cause heat losses.
- Installation is easy and there is no maintenance because each tube can be installed independently.
- When compare the other flat-plate collectors at the same conditions, evacuated tube collectors give a reduction of collector area around 50%.

6.2.2.1.5. Summary-solar collectors. Different types of solar collectors can be used at various solar-assisted cooling systems. The performance of the various collector types are shown with the different types of thermally driven cooling technology in order to provide optimum selection (Fig. 27)

For example:

- Highly efficient evacuated tube collectors are reasonable for double effect absorption chiller.
- All types of evacuated tube collectors can be used for single effect absorption chiller.
- Flat-plate collectors can be considered for desiccant cooling technology.

6.2.2.2. Hot storage tank. Hot storage tank is another important component in the heat-supply circuit. Hot storage tank can supply store excess solar heat for later usage when solar radiation is insufficient to fire the chiller. Some positive effects of the hot storage tank will be described as follows [61];

- Provide sufficient energy to the chiller.
- Store excess solar heat from fluctuating source for later usage.
- Required heating capacity for the auxiliary heating machine is reduced by the heat storage.
- Decouple the solar collector area and chiller.

6.2.3. Cold supply circuit

The cold supply subsystem primarily consists of solar-thermal driven chiller, cooling tower and cold storage tank.

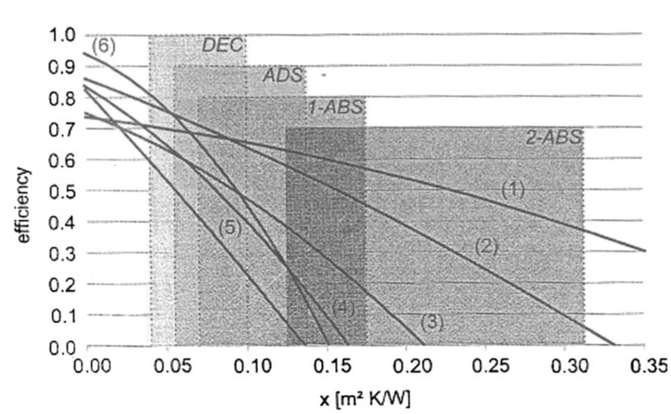

Fig. 27. Comparison of various cooling technology ranges for different types of solar collectors [61].

6.2.3.1. Chiller. Solar thermal driven chillers can be categorized as absorption, adsorption and desiccant cooling systems. In this part, mainly, small size absorption chillers are described in detail with comparative anaysis. In this study, absorption chiller is utilized and the chiller model will be discussed in the following section.

6.2.3.2. Cooling tower. Cooling tower is a heat rejection device which transfers waste heat to the atmosphere. In general, there are two main types of cooling towers available in the market. Those are wet cooling tower (open-circuit) and dry cooling tower (closed-circuit). Wet cooling towers, which are the most preferable one, make use of evaporation with some of water to cool a coolant (see Fig. 28) [64,65].

There are some advantages of the wet cooling tower, for example; it has reasonable heat transfer characterictics and it may provide cooling for the ambients wet-bulb. However, wet cooling tower always requires water, so it is not useful in dry climates.

Unlike wet cooling tower, dry cooling tower causes much more electrical consumption because of its larger fans. Besides this, except dry areas, dry cooling tower has higher investment costs [61].

6.2.3.3. Cold storage tank. Functions of cold store tank are similar to hot storage tank that was described before.

7. Investigated-small size absorption chillers

The development of small-size absorption chillers for taking part on the market seems not easy due to the ruthless competition with electric chillers and heat pumps. On the other hand, it is possible to find five different absorption machines by different manufacturers that could be mentioned more as prototypes. Their features are analysed for the feasibility in commercial use.

7.1. Machine A

Two separate steel units of the evaporator–absorber and generator–condenser pairs are placed, respectively, in this system (Fig. 29). Water/lithium bromide pair is used as the refrigerant–absorbent, respectively. It has developed with a regenerator (plate heat exchanger) between the absorber and the generator for the classic single stage absorption cycle. The circulation between the absorber and the generator is provided by two electric pumps, and a vacuum pump is built-in the system for periodic suction of incondensable gases. The cooling capacity is approximately 11 kW under fixed conditions and a 35 kW evaporative cooling tower is accomplished for the heat extraction under all conditions [66,67].

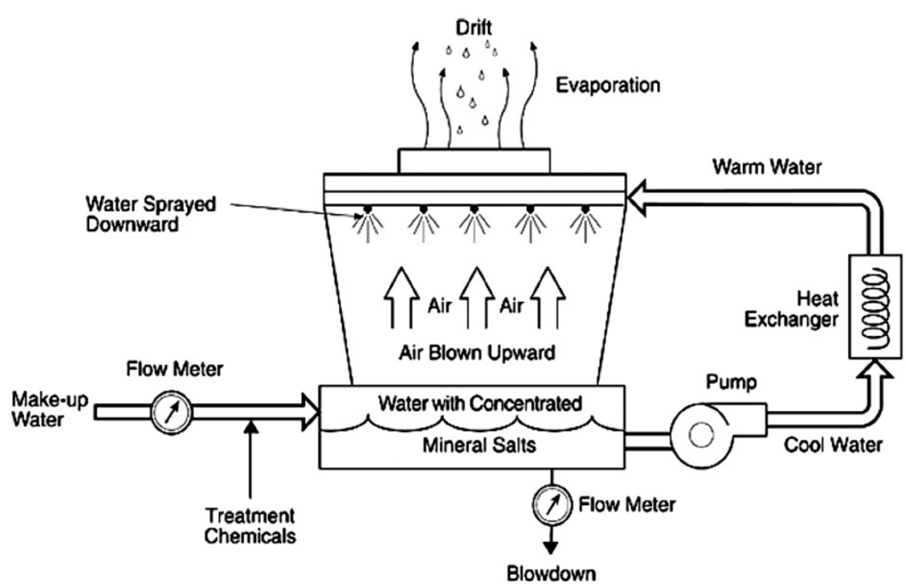

Fig. 28. Schematic diagram of a cooling water system [64].

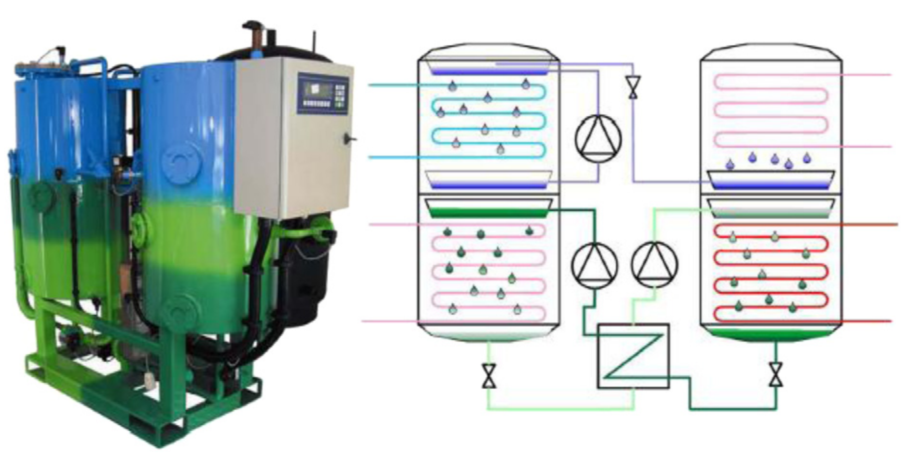

Fig. 29. External and internal view of the machine A [68].

7.2. Machine B

The second examined system has a similar construction with the previous one which follows the normal water/lithium bromide single stage absorption cycle. It is differentiated by only one electric solution pump divided by the heat regenerator to overcome the pressure difference between the absorber and the generator (Fig. 30). Manufacturer test results suggest a wide working range of generator temperatures (from 55 to 105 °C); the cooling capacity is about 10 kW under fixed conditions, and power requirement for evaporative cooling tower is 25 kW [69,70].

7.3. Machine C

All the systems more or less have the same main characteristics, although the difference for this system is the absence of the solution pump. Instead of a solution pump, the system is equipped with a bubble pump for the circulation from the absorber to the generator that does not need electricity. The system uses a plate heat exchanger as a regenerator and operates with a single stage water/lithium bromide absorption cycle. The cooling capacity is 15 kW under the fixed-conditions with coupling requirement to an evaporative cooling tower at 45 kW [14]. This model was basically developed and tested at the University of Perugia labs where a solar field was installed with measurement equipment and a data acquisition system (Fig. 31). The aim of the field is to examine overall functioning of the solar cooling plant under the influence of external circuits. Though this study covers the data provided by the manufacturer, the test results at the University of Perugia for solar-driven system can be found in the listed Ref. [71].

7.4. Machine D

The fourth sample has a rotating generator which is a clear characteristic among the other systems. According to the manufacturer, the generator rotates at a speed at about 4.3 rps in a chamber. This characteristic idea for the generator enhances the heat and mass transfer process itself, permitting a consistent size reduction. Although the cycle is carried out in accordance with normal water/lithium bromide single stage absorption cycle, system established with a wet-type dispersion device (Fig. 32) and the nominal cooling capacity is about 5 kW [72].

7.5. Machine E

The last system (Fig. 33) is considerably different from the traditional absorption processes because a triple-state absorption technology is based on solid, solution and vapour with a water/lithium chloride solution. The triple absorption process works intermittently with two parallel accumulators where each accumulator includes a reactor and a condenser–evaporator. In the charging period, the salt of LiCl was dried for the conversion of the input heat into chemical energy. Subsequently, inversion of the cycle provides cooling effect. There is a requirement for a heat sink for reducing the heat such as evaporative cooling tower in both sequences. The system has nominal cooling capacity about 4 kW [73].

Fig. 30. External and internal view of the machine B [68].

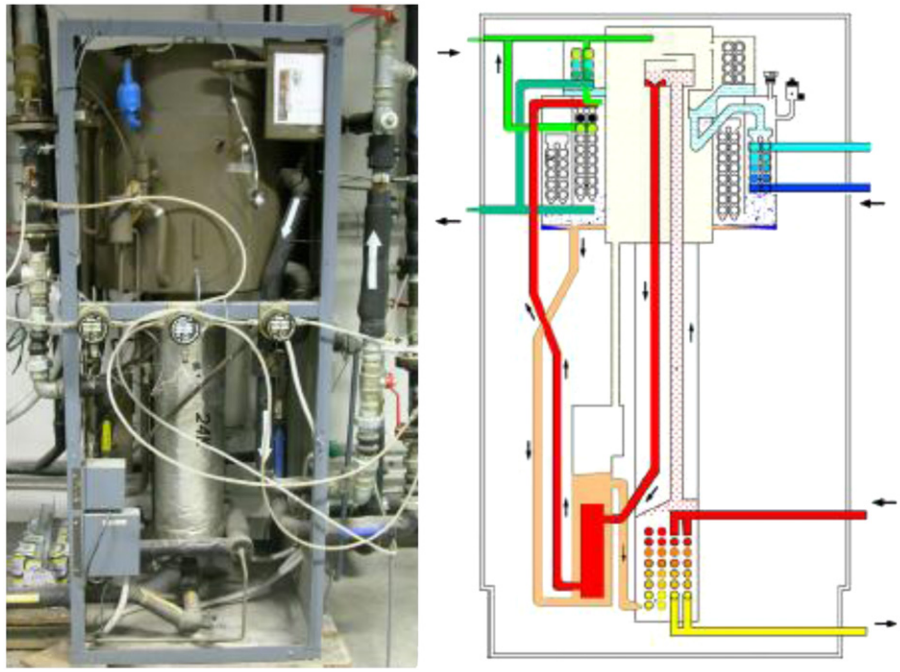

Fig. 31. External and internal view of the machine C [68].

Fig. 32. External view of the machine D [68].

7.6. Other chillers

An increasing attention is attached to small-size absorption machines by researchers and companies and other prototypes are under development such as a single effect ammonia/water absorption chiller equipped with a membrane solution pump [74], providing nominal cooling capacities between 5 and 20 kW. Since the above mentioned systems are still under development, they need to wait for the completion of experiments and evaluation of experimental data. They only stated as the reported performance values in nominal conditions for the comparison with the other five samples.

7.7. Comparative analysis

The five different absorption systems compared and the evaluations were based on cooling capacity and global coefficient of performance starting from the manufacturers' rating and functioning curves (the ratio between cooling capacity and the sum of the heat given to the generator plus the electric energy absorbed). The generator-absorber pumps (when applicable), the pumps for the circulating fluids in the evaporative cooling tower and the solar circuit, the evaporative cooling tower engine and the generator rotation in sample D are the components of the systems that require electric.

The energy consumption values were deemed to be 20 W/kW for fluids transportation in nominal conditions by the external circuits pumps (regarding a direct connection between solar collectors and absorption machine, without cold storage), and 10 W/kW by the evaporative cooling tower engine.

The results for cooling devices that were powered by solar collectors are summarized in Table 18; the values were set as follows:

- Generator inlet temperature $T_{g,i}=85\ ^{\circ}$C.
- Machine outlet cooling fluid temperature $T_{c,o}=9\ ^{\circ}$C.
- Mevaporative cooling tower outlet re-cooling fluid temperature $T_{t,o}=30\ ^{\circ}$C.

The overall dimensions and the weight of each absorption machine together with the unitary cooling capacity were also given in the table. (The ratio between the nominal cooling capacity and the volume of the parallelepiped circumscribed about the system). The small-size absorption machine targeted mainly residential applications where system volume could represent an important feature. Therefore, the volume was chosen as the normalization parameter.

By altering the three external inlet temperatures, the COP and the normalized cooling capacity were evaluated consequently for a more in-depth comparison. The following hypothesis is presumed if data are not directly available:

The variation of the cooling capacity with the chilled water temperature can be plotted as the cooling capacity trend vs. the re-cooling water temperature at a fixed cooling water temperature and the cooling capacity from the scaled graph could be determined at another re-cooling water temperature.

Fig. 33. External view and cross secrion of the machine E [68].

Table 18
Comparison of the characteristics of the investigated absorption chiller machines [68].

Position	Parameter	M.U.	Machine A	Machine B	Machine C	Machine D	Machine E	Machine F
Cooling Circuit	Capacity	kW	11.4	8.4	9.3	5.2	4.8	10
	$T_{c,o}$	°C	9.0	9.0	9.0	9.0	9.0	16
Heating circuit	Power	kW	16.6	13.0	16.0	7.6	7.1	15.6
	$T_{g,i}$	°C	85	85	85	85	85	75
Recooling circuit	Power	kW	28	21.4	25.3	12.8	11.9	25.6
	$T_{t,o}$	°C	30	30	30	30	30	24
Electric absorption		kW	1.4	1.1	1.8	0.7	0.7	1.3
Global COP		–	0.64	0.59	0.52	0.63	0.62	0.59
Weight		kg	700	350	325	240	740	350
Overall dimensions	Length	mm	1500	855	750	1130	700	800
	Width	mm	750	653	716	720	680	600
	Height	mm	1600	1847	1750	790	1850	2200
Volume		m^3	1.800	1.031	0.939	0.643	0.881	1.056
Norm cooling capacity		kW/m^3	6.4	8.1	9.9	8.1	4.8	9.5

In Figs. 34 and 35 the normalized cooling capacity and the global COP are sketched, respectively, as a function of the cooling circuit outlet temperature, setting the generator inlet temperature at 85 °C and the evaporative cooling tower outlet re-cooling fluid temperature at 30 °C.

The first group of graphs point out that the global COP of samples A, B, D and E show weak variations with the cooling circuit outlet temperature that is very close to each other. Sample C proves to be the most powerful machine varying $T_{c,i}$ but it suffers in terms of performance.

In Figs. 36 and 37, by settling the cooling circuit outlet temperature at 11 °C and the evaporative cooling tower outlet re-cooling fluid temperature at 30 °C, the normalized cooling capacity and the global COP plotted, respectively, as a function of the generator inlet temperature. The graphs indicate the same considerations for the $T_{g,i}$ variation for sample C; the normalized cooling capacity of samples B and D is higher than the samples A and E, whereas sample E shows the weakest performance. It is

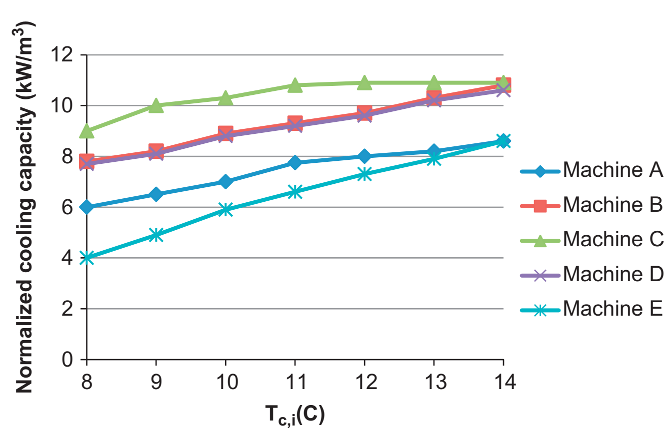

Fig. 34. Machines' normalized cooling capacity as a function of $T_{c,i}$ ($T_{t,o}$ = 30 °C, $T_{g,i}$ = 85 °C), [68].

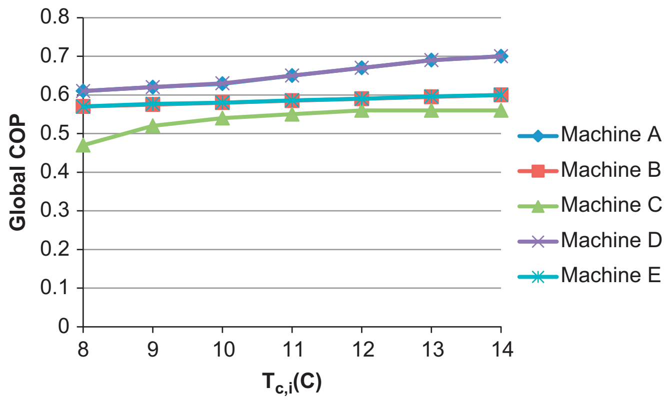

Fig. 35. Machines' COP as a function of $T_{c,i}$ ($T_{t,o}=30$ °C, $T_{g,i}=85$ °C), [68].

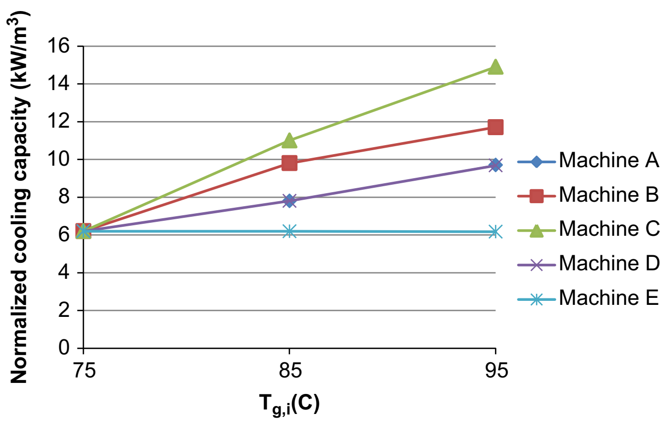

Fig. 36. Machines' normalized cooling capacity as a function of $T_{g,i}$ ($T_{t,o}=30$ °C, $T_{c,i}=11$ °C), [68].

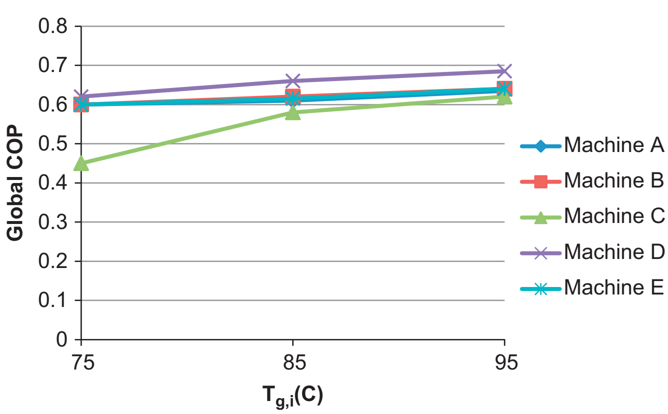

Fig. 37. Machines' COP as a function of $T_{g,i}$ ($T_{t,o}=30$ °C, $T_{c,i}=11$ °C), [68].

shown that the global COP was hardly influenced by the generator inlet temperature.

In Figs. 38 and 39, the normalized cooling capacity and the global COP are, respectively, plotted as a function of the evaporative cooling tower outlet re-cooling fluid temperature where temperature values settled for the cooling circuit outlet temperature at 11 °C and for the generator inlet temperature at 85 °C. The five system particularly samples C and D show bad performance at re-cooling temperatures higher than 35 °C. The differences

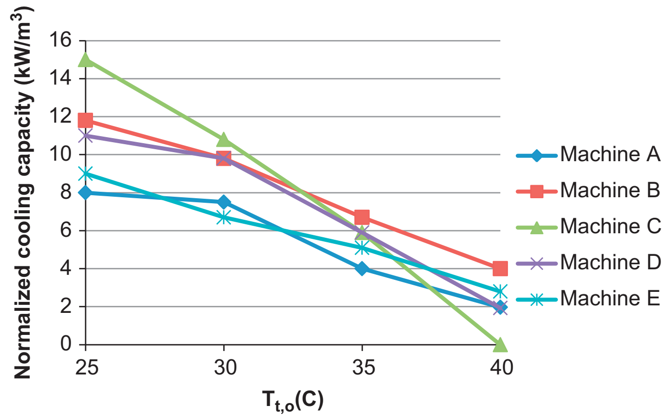

Fig. 38. Machines' normalized cooling capacity as a function of $T_{t,o}$ ($T_{g,i}=85$ °C, $T_{c,i}=11$ °C), [68].

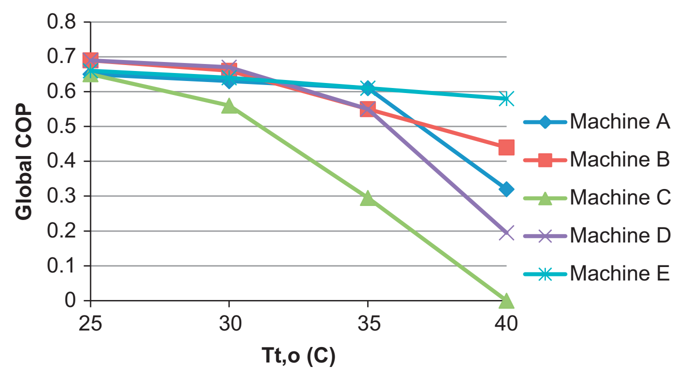

Fig. 39. Machines' COP as a function of $T_{t,o}$ ($T_{g,i}=85$ °C, $T_{c,i}=11$ °C), [68].

between the cooling capacities of the samples depend on the manufacturers' construction choices.

The sample volume significantly weakens the relative capacity for samples A and E. The performance of sample E is the poorest at low cooling temperatures because of decoupling of the heat feeding and cooling production environments. In terms of global COP, samples A and B behave similarly as a result of their common construction philosophy. Additionally, it should be mentioned that samples A and E are two-fold heavier than all the other absorption machines investigated.

7.8. First results of an experimental plant

The preliminary experimental results for the previously mentioned plant installed by the University of Perugia to feed an absorption chiller (D) with solar energy are displayed below.

The results highlighted the effect of generator and cooling inlet temperature effect on the machine performance. A generator inlet temperature of 80 °C producing water chilled at 10–12 °C with a COP of almost 0.6 if cooling inlet temperature is less than 35 °C. When cooling inlet temperature decreases below 30 °C, the chiller cools water down to 7–8 °C, however, COP becomes lower than 0.5. The generator is able to receive more heat than nominal one if the cooling inlet temperature in fact is lower than 30 °C, despite the evaporator use only nominal heat to produce cooling power (Fig. 40). Extra heat is bypassed directly to the absorber in liquid form (overflow effect). As a consequence, preliminary results indicate the feasible working conditions for this machine with a generator inlet temperature lower than the nominal temperature (90 °C), if cooling inlet temperature is over 30 °C with a COP of

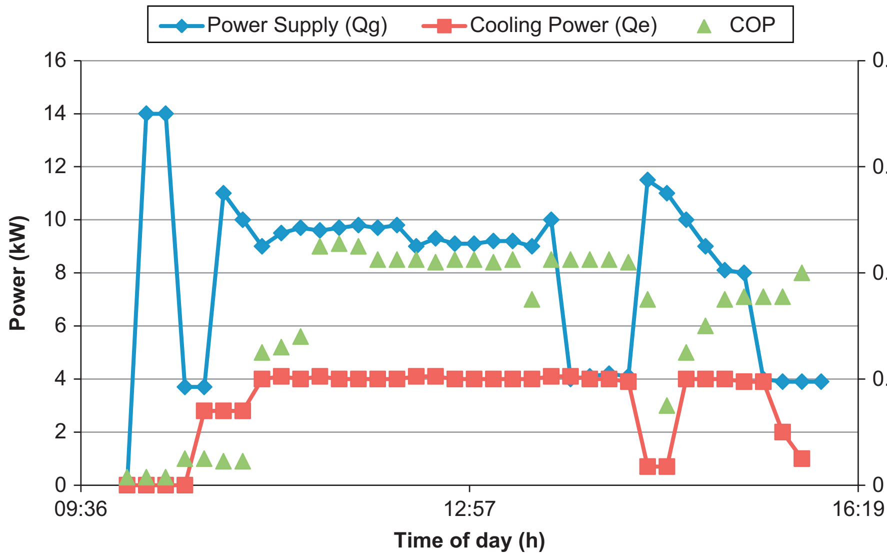

Fig. 40. COP, Power supply (Q_g) and cooling power (Q_e) of Machine D ($T_{g,i}$=80 °C, $T_{t,o}$=26 °C), [68].

0.5–0.6. Although, lower cooling inlet temperature at the same conditions can decrease COP values below 0.5.

7.9. *Conclusions—small scale absorption chillers*

Solar energy based small-size absorption chillers were investigated with five machines differentiated with cooling capacities (from 5 to 12 kW) using dissimilar characteristic working principles and designs. The performance study was led from the manufacturers' rating and functioning curves, followed by setting the same working conditions for all machines and testing their performance using the solar energy as the heat source. The study has been applied taking into account of the dimensions of each sample and accompanied altering the external temperatures from heat source, re-cooling water and cooling water.

The results consist of the data from the cooling load, the external environment, and the sun impose as the working conditions that carried on with five machines; and allow the characterization of the five samples' performance.

8. Design of a small scale solar thermal absorption cooling system

There was a great interest in research and development of air-conditioning (cooling) systems that use solar power like solar powered absorption cooling systems. By 1960s, LiBr/H2O usage was reported for absorption refrigeration processes as a new technological innovation by many research groups including Chinnappa [91,92], Duffie and Sheridan [93], Sargent and Beckman [94], Perry [95], Nakahara et al. [96], Auh [97], Doering [98], Alizadeh et al. [99], Ward et al. [100,101] and Ward [102].

In the past decade, Grossman and Zaltash [103], Sumathy et al. [76], Pohl et al. [104], Asdrubali and Grignaffini [71], Hiebler et al. [105] and many others have developed and experimentally tested solar air absorption cooling systems that depend on the different principles. Asdrubali and Grignaffini reported the highest coefficient of performance (COP) at temperatures around 70 °C by using of a 17 kW single effect LiBr/H2O absorption cooling system powered by a 30 kW electric boiler. The higher the coefficient of performance, the more efficiency in the system is achieved. Also the agreement between numerical and experimental data was shown with a numerical model developed for the simulation of experimental conditions [75].

It can also be stated as the efficiency ratio of the amount of heating or cooling provided by a heating or cooling unit to the energy consumed by the system. Electrical heating for example has a coefficient of performance of 1.0 however; electrical energy consumption was not reported to help evaluation of the electrical COP. An integrated solar cooling and heating system with two-stage absorption chiller (with cooling capacity of 100 kW) was studied by Sumathy et al. in whose project achievement of a COP of between 0.388 and 0.437 was reported by hot water temperatures ranging from 60 to 75 °C, [76]. To make the system efficient and cost effective, it was offered to integrate the cooling system with existing conventional domestic hot water systems again by the same group. Although a large potential market exists for building air-conditioning systems (with less than 10 kW capacity), solar cooling systems to serve domestic housing needs are lacking efficient and economical heat exchangers. Current cooling systems run on electricity and lead to large energy consumption whereas, thermally driven absorption heat pumps offer freedom from electricity supply because they can be powered from the combustion of natural gas and solar and waste heat.

A solar thermal absorption cooling system with a cold store was designed to cool a small scale domestic building by the solar thermal absorption cooling system project for the investigation of small solar powered absorption air-conditioning system success. The solar thermal absorption system cooling efficiency, solar array requirement to power the system, consumed electricity by the system, storage capacity per kW of chilled water produced and system improvement possibilities were tested. The system is a sub-project for the integration of renewable energy into existing and new buildings for zero-energy and energy-plus buildings design strategy.

8.1. *System design and procedure*

The solar thermal absorption cooling system was designed in 2011 to supply cooling to a small domestic building. A schematic diagram of the design of a solar thermal absorption cooling system is shown in Fig. 41.

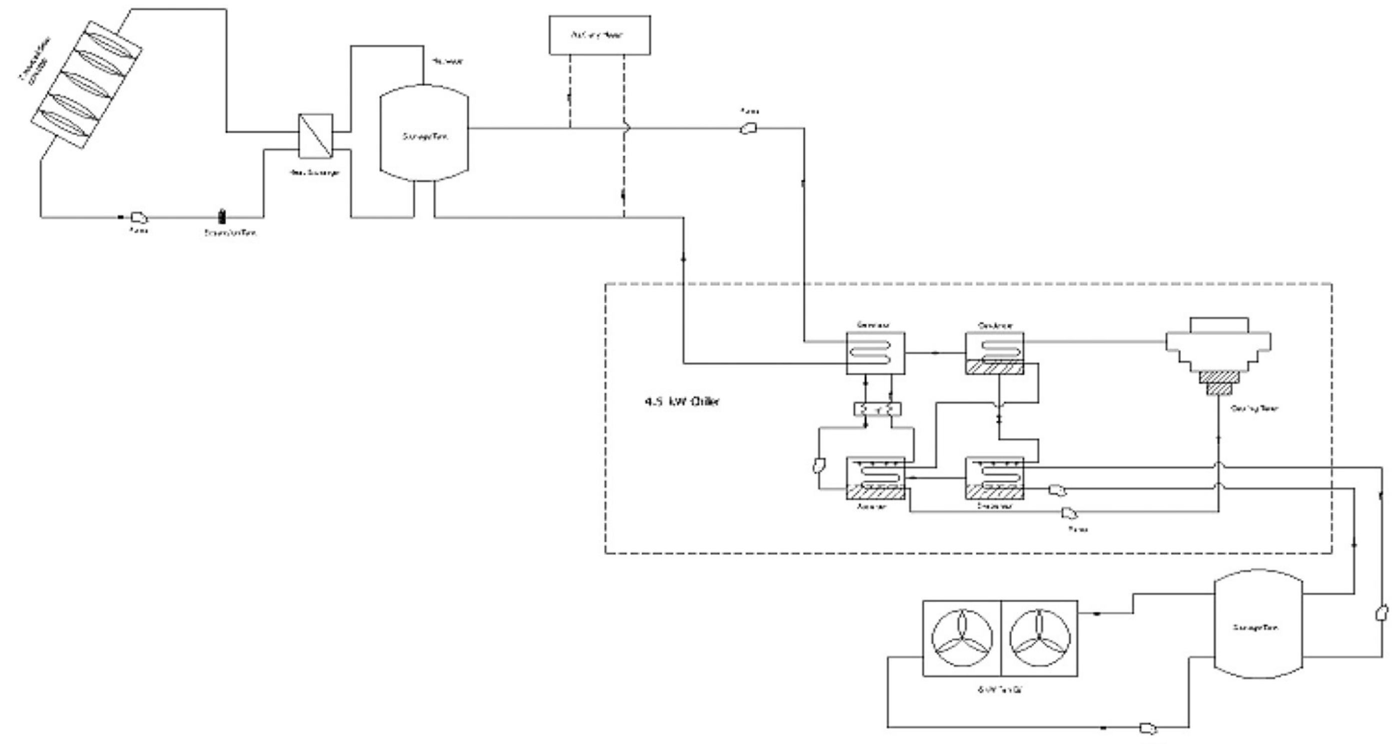

Fig. 41. Schematic diagram of the design of a solar thermal absorption cooling system.

Fig. 42. (a) Evacuated tube collectors [77], (b) Storage tank and Rotartica chiller (4.5 kW), [78,79], (c) Fan Coil (6 kW), [80].

The main components of the small scale solar absorption cooling system were:

- A 15 m^2 (effective) evacuated tube collector, mounted on the roof.
- A 40 kW heat exchanger, transferring heat from collector loop to the chiller working fluid (water) to drive the chiller.
- A 4.5 kW LiBr/H_2O Rotartica semi-commercial single effect absorption chiller, fired by hot water to produce chilled water.
- A 1000 L cold water tank which functions as a buffer between the chiller and the cooling demand from the room.
- A 6 kW fan coil unit to circulate chilled water from the cold store, supplying comfort cooling type air-conditioning to the room.

8.1.1. The collector area

The evacuated tube collector was used for the source of heat for the chiller and mounted side by side with the chiller and the water storage tank on the roof of the air-conditioned room as shown in Fig. 42a and b, respectively.

The collector field consisted of five 30-tube evacuated tube collectors with a net absorber area of 15 m^2 (20 m^2 gross area), facing south with 45^o inclination, [81]. Collector loops at switch-on and switch-off temperatures of 4 and 2.5 °C between the collector inlet and outlet fluid temperatures controlled by a controller. Minimum switch on fluid temperature was set at 50 °C. At a switch-on temperature difference of 4 and 50 °C fluid temperature, the collector cycle pump and the chiller feed pump

were switched on simultaneously and switched off simultaneously with a temperature difference of 2.5 °C. The chiller generator was fed with input heat by exchanging the collected energy from collector fluid with water through a 40 kW heat exchanger.

8.1.2. Rotartica chiller (4.5 kW)

A minimum temperature of 80 °C was used for activation of the Rotartica chiller unit. The chiller employed a LiBr/H_2O pair as a refrigerant. Through providing heat for the chiller generator, salt along with the refrigerant (water) or absorbent (LiBr) within the generator vaporized. LiBr in solution form was conveyed to the absorber with low refrigerant content while evaporation process distillated the water from the solution in the generator flowed to the condenser where it was condensed to release heat. The heat can have diverse sources, such as, solar thermal collectors, cogeneration systems, boilers and other residual heat sources the difference in pressure caused a flow of the refrigerant to the evaporator where, it evaporated and absorbed heat that subsequently produced chilled water.

As the temperature and pressure was low at evaporator, the refrigerant flew through the evaporator because of difference in pressure. The refrigerant evaporated and absorbed heat that subsequently produced chilled water. The absorbent in the absorber attracted the evaporated refrigerant and formed the refrigerant rich absorption solution. Then the solution was conveyed to the generator to start the whole cycle again. If the temperature dropped below 70 °C for the feeding water of the chiller, the absorption process in the chiller stopped. However, even absorption process stopped in the chiller, the chiller itself continued to run and consume electrical power. Produced cold water at a temperature of between 7 and 16 °C was transferred to be stored in a 1000 L cold water tank. Input for chilled water placed at the bottom of tank and warm water output at the top of the tank. The temperature distribution in the tank could be measured by resistance temperature detector sensors and the average was taken to determine the temperature of bulk store.

Cold water from the bottom of the store tank was pumped to a 6 kW nominal cooling capacity fan coil to cool an 80 m^3 building. The operation of the fan coil was controlled by installing a thermostat in the office that was set to an average temperature of 16 °C. The cycled water from fan coil entered the store tank at the top. Fig. 2 shows the pictures of the core components in the solar thermal absorption air-conditioning system.

8.1.3. Component dimensioning

The cooling demand of a post 2000 three bedroom house (77.3 m^2) with the estimation at 1472 kW h with a peak of 2.1 kW in the summer months in Cardiff, UK was the selection criteria for 4.5 kW Rotartica unit [82,83]. The periods for solar energy would not be available and the lack of small scale solar powered air-conditioners in the market were the other driven forces for the selection. The system configuration was built on the cooling capacity of the 4.5 kW semi-commercial chiller with the optimum COP = 0.7 (advised by the manufacturer) of the chiller.

The required input power to produce cooling capacity of 4.5 kW was expected as 6.428 kW using the following equation:

$$\text{COP of chiller}(0.7) = \frac{\text{Cooling power}(4.5\text{ kW})}{\text{Input thermal power in kW}} \tag{8.1}$$

The evacuated tube collector with a design to cover 100% of the thermal energy required was the thermal energy source for the chiller. The equations are given for the calculations of collector efficiency and performance by the manufacturer as given in Eq. (8.2) and details of the conversion factor and loss coefficients are given in Table 19.

$$\eta(T_m) = \eta_0 - a_1 T_m - a_2 G T_m^2 \tag{8.2}$$

where $T_m = (t_m - t_a)/G$ and $t_m = (t_{in} + t_{out})/2$.

The numerical values from the calculations are listed as 0.64 for collector conversion efficiency and 0.9 with heat exchanger efficiency, presumed 800 W/m^2 levels of incident solar radiation, estimated nominal cooling power of the chiller for the optimal collector size to be 14 m^2. Possible heat losses in the system were accounted by the selection of 15 m^2 of collector. Peak cooling load was estimated of 2.1 kW, henceforth valuations for the capacity of cold water store was based on this estimation (for the extra 2.4 kW cooling produced). An assumption on water cooling was done from 18 to 7 °C for the cooling demand between 17:00 and 22:00 while solar energy would not be available. Since one of the standard tank capacity in the market was 1000 L, estimation for tank capacity was taken at 935 L. The parameters of the tank was listed in Table 20 with parameters, chilled and warm water input and output locations, distribution of the temperature sensors immersed into the tank to measure average store temperature. The appropriate fan coil capacity was settled of 1.2 times the nominal chiller capacity according to the chiller manufacturer.

8.1.4. Measuement of the operational parameters for the system

The system performance was evaluated via an installed data-acquisition system to measure the operational parameters of the system. Also some other components were integrated to the data-acquisition system as resistance temperature detector sensors (0.3% accuracy) placed in various locations in the system, flow meters [85], (1% accuracy) to measure the mass flow rate of fluids in the heat transfer loops, and pyranometer (accuracy 3%) to measure the global solar irradiation received in the plane of the solar collectors. The measurement of the ambient temperature was carried out via two temperature detectors with Stevenson (thermometer) screens, [86]. A data logger system for recording the live data and transmission to the internet was used. The physical presence on the roof which could also effect the efficiency of the system and experimental results was eliminated by online monitoring the system performance remotely.

The energy flow measurement in the each heat transfer loops consists of the chiller inlet loop (between the heat exchanger and

Table 19
Details of the conversion factor and loss coefficients [84].

Parameter	η_0	a_1 $(W/m^2\,K)$	a_2 $(W/m^2\,K^2)$	$G(W/m^2)$	$T_m(^\circ C)$
Value	0.779	1.07	0.0135	800	20

Table 20
Parameters of the tank with chilled and warm water input and output locations [84].

Parameter	**Value**
Tank height	2.058 m
Tank diameter (without insulation)	0.79
Insulation thickness	85 mm
Weight of store (fully filled)	1116 kg
Sensor location(from bottom)	0.310 m, 0.745 m,1.250 m and 1.710 m
Chilled water to store and cold water to fan coil in opposite direction	0.310 m from bottom
Warm water to store and warm water to chiller in opppsite direction	1.710 m from bottom

the Rotartica chiller), the solar loop, the chiller output loop (between the chiller and the cold store) and the fan coil loop (between the cold water store and the fan coil) was carried out on the data obtained from the temperature sensors and flow meters.

The integration of the difference in energy input and output over time was evaluated to determine the thermal performance of the system. The value of the measurement of electrical consumption in the Rotartica unit, the pumps and the fan coil unit utilized in the loops (parasitic load) were also given by the experimental programme to determine totally consumed electrical energy in the system and the electrical COP. The measurement time steps could be 5 min using a data logger for the summer months.

8.2. *Result and analyses*

8.2.1. *Electrical consumption and power values*

Some thermal power outputs such as available solar power, which were considered in a hot sunny day between 11.30 h and 13.30 h, were demonstrated in Fig. 43. Furthermore, total electrical power consumption and cooling capacity were presented with available solar radiation in the same figure.

According to the graph, chiller could not produce cooling power during the morning hours until 9.30 h because there was no enough solar radiation to drive chiller. In order to produce chilled water from chiller, solar radiation must reach at least 440 W/m^2 which provides to drive temperature for the chiller at about 80 °C.

Average electrical power consumption by pumps, fan coil and chiller was around 1.26 kW. After the 9.30 h, solar radiation increased gradually until 12.30 h and producing peak value reaches 9.75 kW of power (812 W/m^2) at 12.30 h. Increase of solar radiation caused to increase the thermal input power that allowed rising cooling capacity of the chiller. From the figure, at 12.10 h, the highest chiller output was observed as 4.98 kW with chiller thermal COP of 0.63.

Maximum solar radiation to produce cooling was 756 W/m^2 (8.00 kW of power) for the 15 m^2 evacuated tube collector. Once, during the day solar radiation was more than 756 W/m^2, excess thermal energy could not be used to generate cooling. Thus, this excess energy should be stored to improve COP of the solar absorption cooling system.

Table 21 showed some details on production of average thermal power and consumption of electrical power for two different periods; between 9.30–17.00 h and over 24 h. For both periods, total electrical consumption was observed nearly the same. However, produced chilled water by the chiller at the period between 9.30 and 17.00 h was approximately three times greater than produced cooling power for 24 h period. Therefore, controlling of the chiller properly will improve the COP of the system and help to reduce the total electrical consumption.

8.2.2. *COP of the solar thermal absorption cooling system*

The general purpose of the small scale solar absorption cooling technology was to understand how efficiently solar cooling system generated cooling and to reduce the footprint of systems for integration with existing and future domestic buildings. One of the most significant issues for the solar cooling application was its competition with other alternative cooling systems such as conventional vapour compression technology.

Table 21
Comparison of average thermal power and electrical consumption for different periods.

Averages (kW)	9.30–17 h	24 h
Available solar power	7.62	2.96
Thermal power input to Rotartica	6.51	2.19
Cooling power from Rotartica	4.07	1.23
Total electrical consumption	1.20	1.19

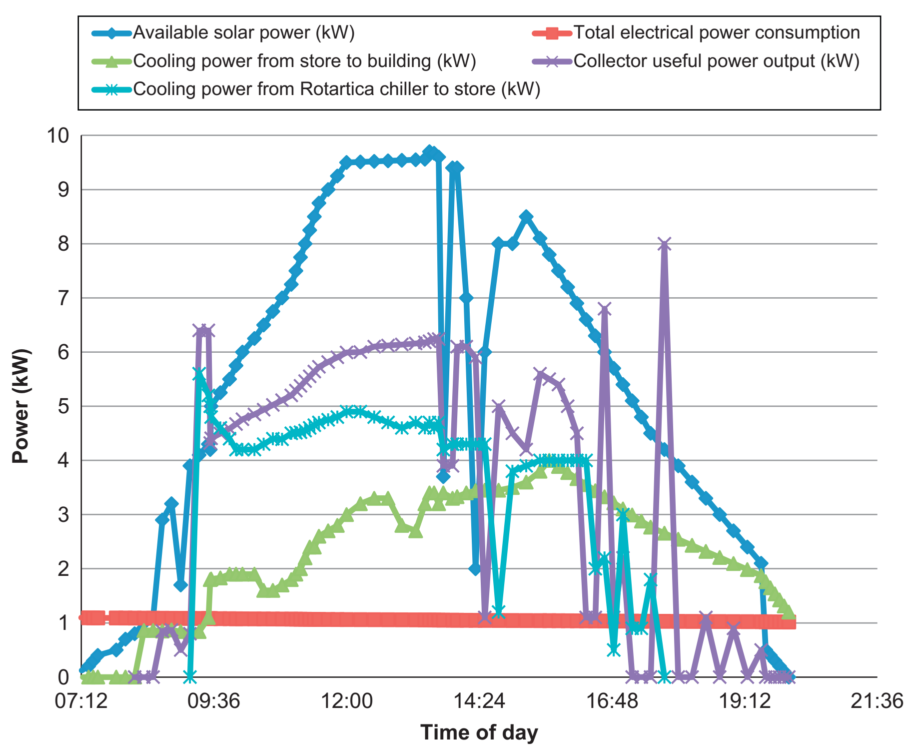

Fig. 43. Available solar power, total electrical consumption and solar cooling capacity of the system in a typical hot sunny day.

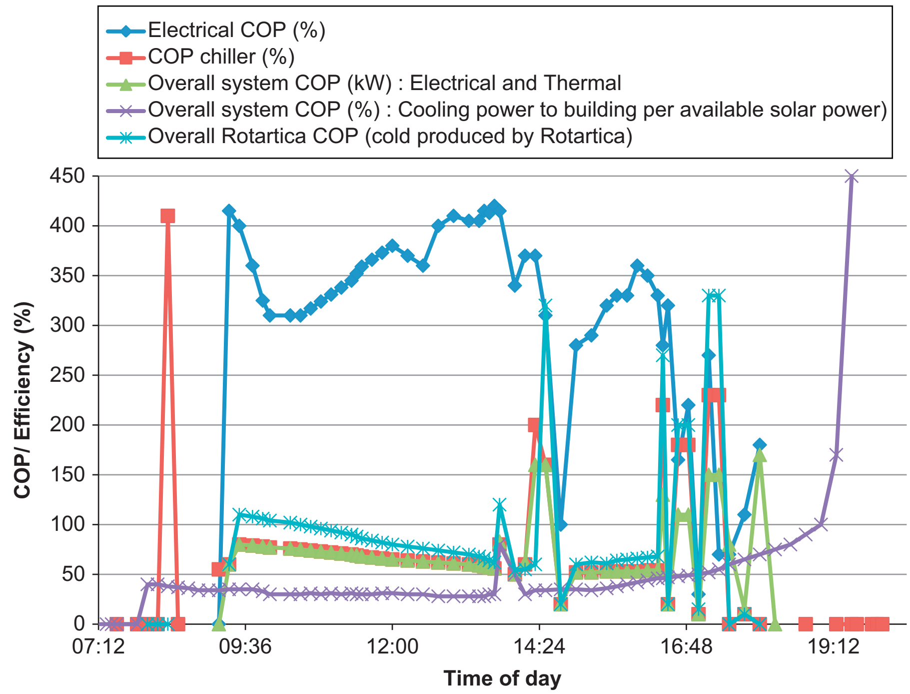

Fig. 44. Performance of the system in a hot sunny day with avarage 800 W/m^2 irridation.

Table 22
Descriptions of the different COPs of the system.

	Description	Average value between 9.30 and 17 h
Cop chiller	Cold power output (thermal) per unit thermal power input to Rotartica unit	0.65
Overall Rotartica COP (thermal)	Cold water power output (thermal) per unit solar power available (thermal) to the collector	0.57
Overall system COP (thermal+electrical)	Cold water power output (thermal) per unit solar power available to collector (thermal) and also electrical power consumed	0.46
Electrical COP	Cold water power output (thermal) by Rotartica per unit electrical power consumption in the solar thermal absorption cooling system	3.62

The following outputs related to the efficiencies were shown in Fig. 44. The efficiencies were obtained with using 15 m^2 effective evacuated tube collector for the conversion of solar energy and 4.5 kW Rotartica chiller to produce cold water, 1000 L cold water tank to store cold water and 80 m^3 building for cooling. Fig. 44 demonstrated performance of solar absorption cooling plant on a typical hot day. The descriptions of the different COPs of the system were shown in Table 22.

The chiller COP was calculated around 0.66 which is very close to ideal value of 0.7. Moreover, driving heat temperature for this single effect LiBr–H_2O absorption cooling system was taken 80–100 °C, [61]. In general, chiller COP considers solar collector efficiency and also, heat exchanger gives a lower average value around 58%. Furthermore, the chiller, pumps and fan coils cause a far lower value for whole system COP of 47%. Although, an average electrical COP was measured about 3.64 that seem good, it is possible to improve the average value around 5 with some improvements on Rotartica chiller unit.

In comparison to other studies in the same conditions, the results of this study showed similar outputs. For example; a long-term monitoring research [87] on four liquid and direct expansion mechanical vapour compression chillers has achieved total system COP (thermal output, electrical input) of around 0.8 up to 1.7 with a maximum monitored daily COP of about 3.4 for the small direct-expansion (DX) split where water distribution was contained COP figures that are nearly the same or a bit of worse than those succeeded by the this small scale solar thermal absorption cooling system. This means that, once components of solar thermal absorption cooling system have been optimised, the system will be capable of better world technology.

8.2.3. Analyses of electrical power consumption for the small scale solar thermal absorption cooling system

Analysis of electrical power consumption for the system was shown in Fig. 45. It can be clearly seen that chiller consumed major electrical power, accounting around 82% of the overall power consumption in the system. The chiller has some components which consumed electrical power; for example; according to the manufacturer data, while fan consumed around 300–450 W, rotary unit consumption was around 300 W electrical power. However, there was no idea about pumps electrical power consumption.

It is possible to reduce electrical power consumption which is caused by chiller. For instance; especially during the night, there was no enough heat energy supplied to drive the chiller and to

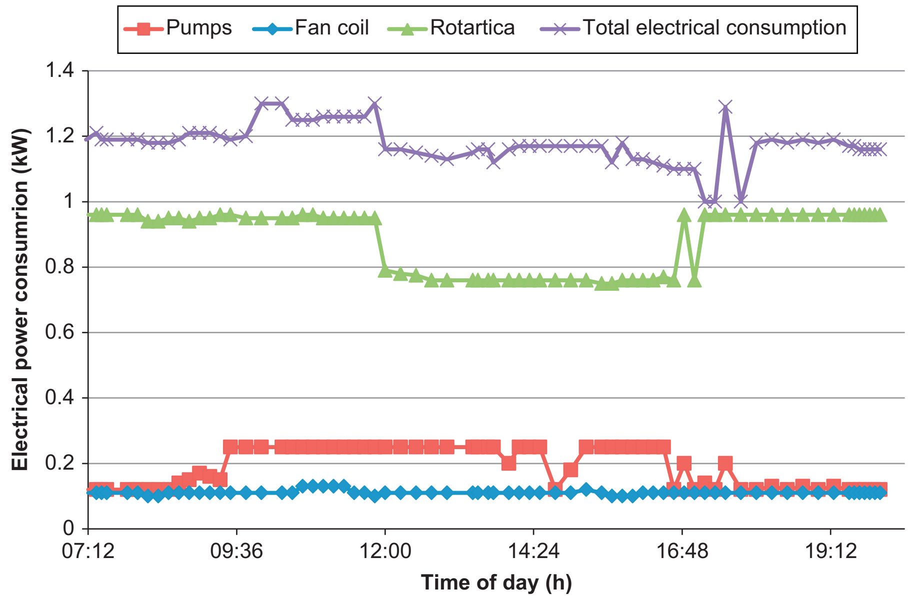

Fig. 45. Analysis of electrical power consumption of the system in a hot sunny day.

produce the chilled water output but the chiller continued to run. Unexpected electrical power consumption was occurred because of running with no chilled water output. Once the chiller is controlled properly, it will be possible to reduce electrical power consumption.

Chiller inlet temperature and collector should be turn on when the output temperature reaches the around 50 °C. According to Fig. 45, overall electrical power consumption increased from 1.2 kW to 1.3 kW between the period of 9.20 and 9.35 h. The reason of the rise was that pumps started to work in the collector and chiller inlet cycle. At this point, rising of electrical consumption showed the pumps consumed from 0.12 to 0.22 kW. Total electrical power consumption remained steady at 1.30 kW until 12.30 h. After that, total electrical consumption decreased from 1.30 to 1.17 kW because of reduction of power consumption from chiller fan. When produced cold water temperature fell to below 9.5 °C, chiller unit power consumption reduced from 0.97 to 0.84 kW. This shows the total electrical power reduction approximately 10.0%.

8.2.4. *Temperature readings*

At this section, the time variation in various temperature such as ambient temperature, chilled water temperature, chiller input temperature and collector output temperature was demonstrated for a typical sunny day (812 W/m^2 peak solar radiation). (see Fig. 46).

It was expected that the solar radiation measurements increased from the morning at around 7.00 to 13.30 h and then decreased regularly to zero at about 20.00 h. As expected, high solar radiation period was observed similar with the ambient temperature between 11.30 and 16.00 h when cooling might be mostly required. The figure also showed that rising in solar radiation during the day caused the increasing of collector outlet and chiller inlet temperature. When initial chiller inlet temperature was around 80 °C, absorption process started to generate chilled water. After at 10.00 h, energy requirement was possible to the chiller for start-up due to the suddenly reduction in chiller inlet and collector output temperature.

Nevertheless, with increasing solar radiation, the collector output and chiller inlet temperature grew up gradually and reached their maximum values of 93 and 88 °C, respectively, at 12.30 h. The temperature of chilled water produced continued to decrease to at around 7.4 °C that is a minimum value. Furthermore, some average performance details were given for various operating conditions (see Table 23). The performance values were measured for a typical hot day between the time 9.30 and 17.00 h.

Generally, the availability of solar radiation influences the performance of the solar absorption cooling plant directly. At the system condition, 4.1 kW cooling power could be produced between the period of 9.30 and 17.00 h. Requirement for cooling of 80 m^3 building was around 2.82 kW with a 1.27 kW storage capacity. Once, that cooling demand was considered for the system, aim of the storage temperature might be 7.0 and 10.0 °C during the day. In addition to this, when the excess solar radiation was succeeded, the system will require between 180 and 250 L storage for per kW of chilled water generation.

Also, the application required between 60 and 90 L storage for cold store each square meter of evacuated tube collector. When the cooling requirement and overall available solar power for the collector was considered, total system thermal efficiency was approximately 40%. Some improvements will be done to increase reliability of the system. For instance; as it was mentioned before, solar radiation might be not enough to drive the chiller particularly during night and overcast condition. It is possible to prevent this drawback with some optimisation on the system such as changing material phase or providing heat requirements from LPG to drive the chiller.

8.2.5. *System performance*

Production of cooling power from solar absorption system for various overall solar radiations in the summer was presented in Fig. 47. The achievable ranges of cooling power for the different solar radiation levels were also marked. The increase of solar radiation on the collector surface led to increased cooling power. When the irradiation increased up to 245 W/m^2, maximum 1.3 kW cooling water power was generated by the system.

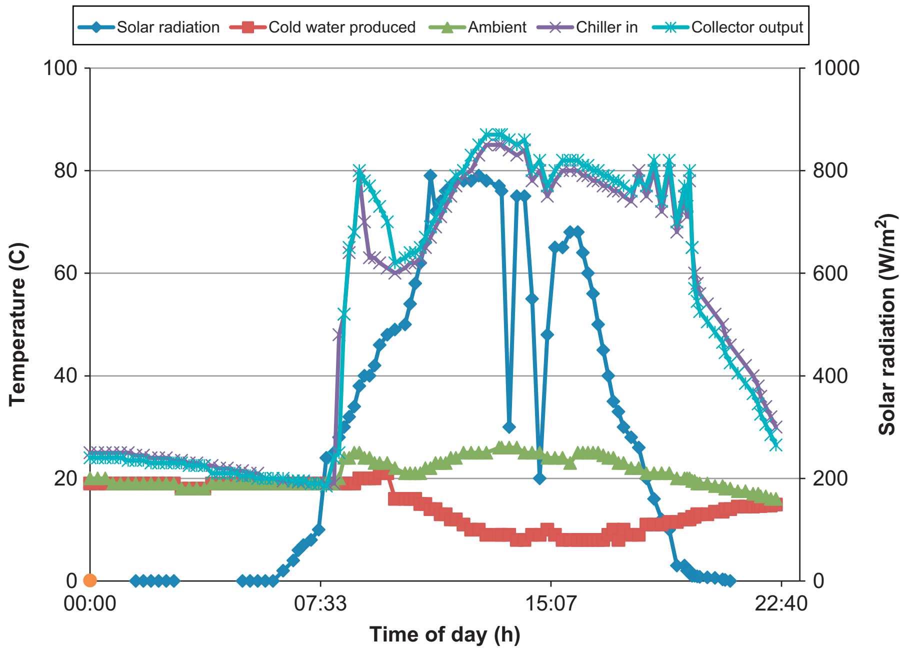

Fig. 46. Time variation of various temperatures in a typical hot sunny day with irradiation of 812 W/m^2.

Table 23
Some average performance details for various operating conditions in a typical hot sunny day with 812 W/m^2 peak solar irradiation.

Chiller inlet temperature (°C)	70	75	80	85.3	88	78[a]
Solar irradiation (W/m^2)	617	712	760	791	800	632
Ambient temperature (°C)	22	22.6	23.3	24.5	24	23.8
Collector output temperature (°C)	74	79	84	89	92	80
Chiller return (hot side) temperature (°C)	62	66.1	70.3	76	78.2	69
Chiller inlet temperature (cold side) (°C)	16.5	15.5	14.5	13.2	12	13.8
Chilled water temperature (°C)	12.2	11.2	10	8.8	7.3	10
Average store temperature (°C)	15.4	14.3	13.1	12	10.8	12.6
Fan coil inlet temperature (°C)	13	12.3	10.8	9.6	8.4	10.4
Fan coil outlet temperature (°C)	15.8	15.2	14.4	13	12.6	14
Cooling power to fan coil (kW)	2.28	2.45	2.90	2.65	3.20	2.79
Chiller COP	0.66	0.64	0.63	0.60	0.60	0.66
Electrical COP	3.5	3.6	3.78	4.0	4.2	3.6

[a] Average values between 9.30 and 17 h.

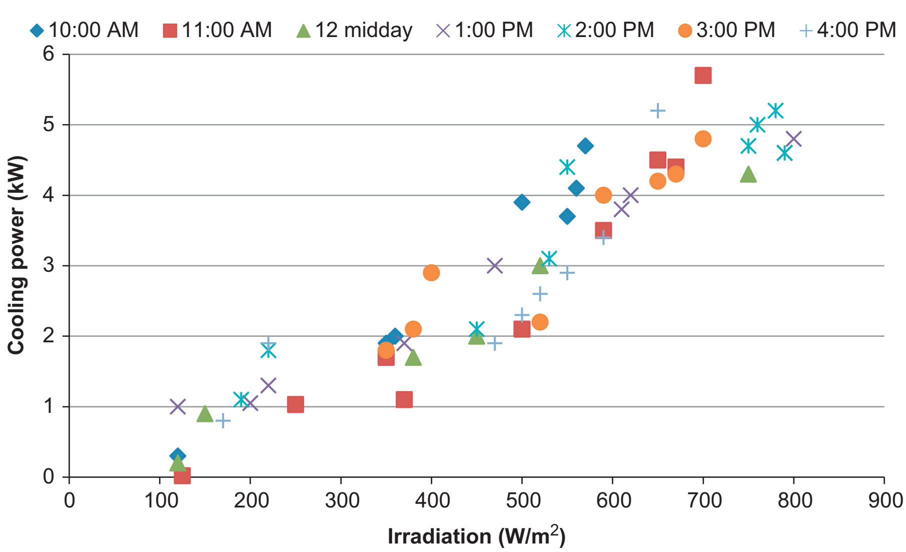

Fig. 47. System performance for different solar irradiation levels.

At this condition, an electrical COP in the system is equal to 1.0 because produced cooling power (1.3 kW) was equal to the average electrical consumption of the application. As it was seen in the graph, in order to reach the demand of cooling power output, minimum insolation level must be minimum 580 W/m^2. Furthermore, once insolation level reached more than 756 W/m^2 with the 15 m^2 evacuated tube collector, excess solar heat could be stored for later use.

9. Future work

Further improvement for this study will be integration of the ground-source heat pump (GSHP) in the system. The GSHP will back up for the solar thermal during the cooling. The soil can be used as the heat sink to cool the building directly. For example; GSHP loop can be used as a storage tank to absorb the excess solar energy especially during the summer. This excess solar energy can be utilized when solar radiation is insufficient to fire the system. Moreover, GSHP cycle as utilized storage tank that allows using small scale storage tank for whole system will also help to reduce the footprint of the cooling system. Furthermore, it is expected that the performance of the solar thermal absorption cooling system will increase with additional cooling mode (GSHP).

Ground-source heat pump (GSHP) is an energy utilization technology of high efficiency, energy saving and environmental protection. The appearances of the global energy crisis and the increase in environmental problems caused it's gradually rise. The ground surface heat is considered as the heat source (heat sink) of the heat pump, and the low grade energy is transformed into the high grade energy by inputting a small amount of high grade energy (e.g., electrical energy). The GSHP can be classified into three types in accordance with the definition of ASHRAE: the ground-coupled heat pump (GCHP), the ground water source heat pump (GWHP) and the surface water heat pump (SWHP). As the GCHP has advantages such constant temperature which equals the local average ambient air temperature in the ground 10 m below, less pollution and ease of integration with solar energy; it has gained more attention recently.

The GCHP use for heating often requires a larger ground heat exchanger (GHE) which may be restricted by the construction sites, thus usually combined with solar collectors [88–90].

Generally, the solar collection system will involve as following sub-systems: independly developed high efficiency evacuated tube solar collectors with a total area 20–25 m^2, an expansion tank, a circulating pump and pipelines. The underground heat exchange system contains the GHE, circulating pumps and pipelines. The GHE consists of three vertical single U-tubes and require the instalment below the house with the individual depth of 40 m (see Fig. 48). The U-tubes are usually made from high-density polyethylene (HDPE) and sand is used as the filling material of the boreholes. The water/antifreeze mixture-ethylene glycol solution will be preferred all along in the GHE since it is difficult to replace.

10. Conclusion

There is a great demand for solar thermal air-conditioning system over the world, because these systems have a strong potential for remarkable primary energy savings. Moreover, solar cooling has a wide market choices, so significant cost reduction can be possible in short-medium term.

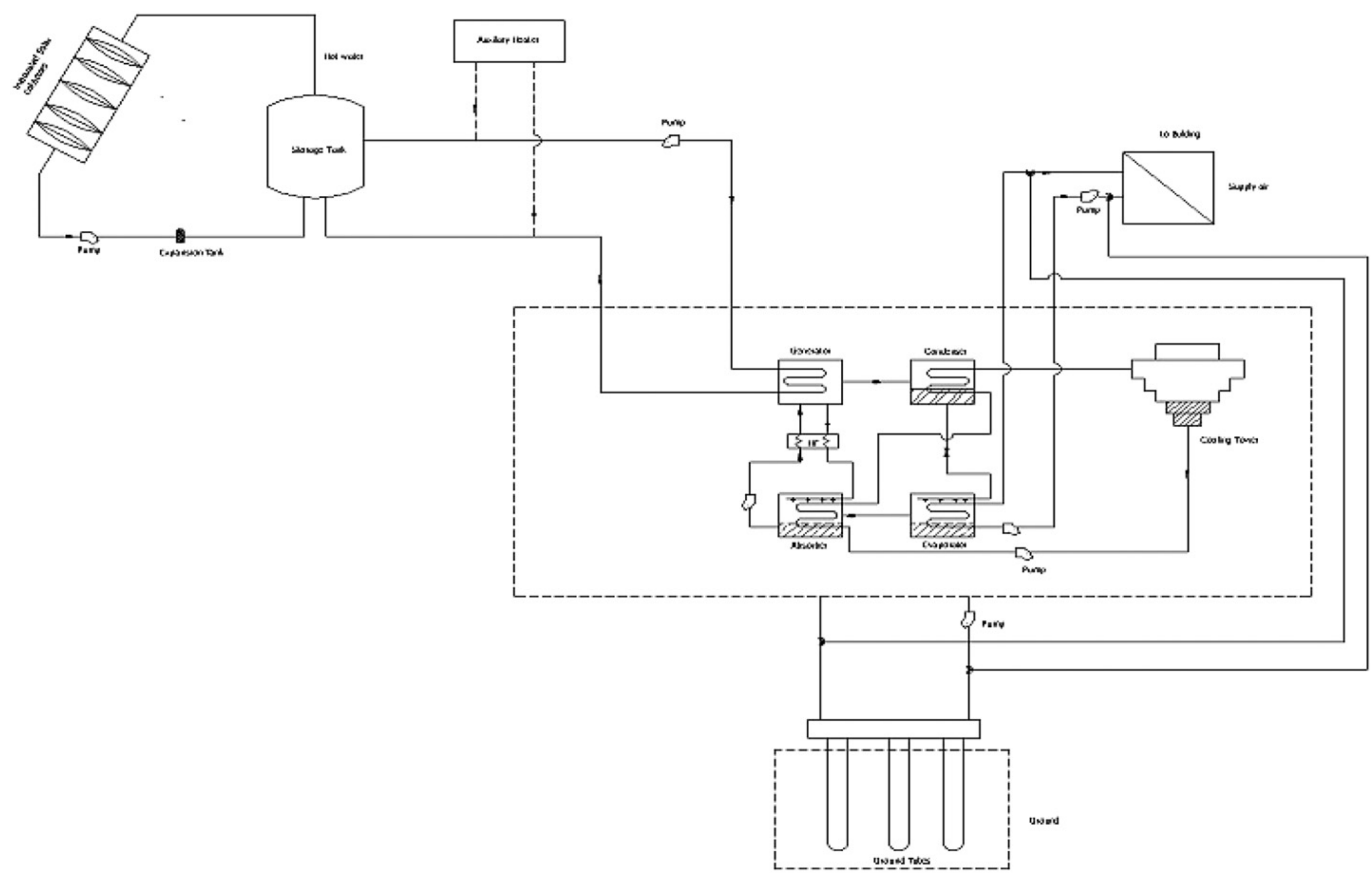

Fig. 48. Schematic diagram of future work solar cooling system.

The main advantages of solar thermal technology, [60];

- Save CO_2 emissions.
- Proven and reliable.
- Save heating/cooling bills.
- Rapid growth technology (available all over Europe).
- Reduce the dependency on imported fuels.
- Save rare natural sources.

Although the solar cooling technology is still in early step of development, today approximately 70 solar-assisted air conditioning systems are established in Europe. Particularly, in last 5 years, researches and developments with novel installations have been increasing rapidly, some of them were given above as examples and new developments. Probably, thermally driven chillers and desiccant cooling system will play a key role in approaching economic feasibility. New development activities are necessary in order to promote market integration and to reduce the cost of using solar-thermal air conditioning in buildings. Such new systems will be a future option for sunny climates zone.

The aim of this project was reducing the footprint of solar thermal air conditioning system. For this reason, a small scale solar assisted LiBr/H_2O absorption air-conditioning system was designed with some improvements. In fact, this study shows how solar energy could be used to generate clean cooling in domestic buildings. Average chiller inlet temperature was calculated approximately 78 °C and total electrical COP of 3.64 was achieved for the application. The average ambient temperature is 24 °C and also, in a hot sunny day (insolation 812 W/m^2) average thermal COP was calculated around 0.58. When excess solar radiation is succeed, additional storage between 180 and 250 L may be required for each kW of cooled water. Moreover, for each square meter of evacuated tube collector, also storage between 60 and 90 L may be required. Today, unfortunately this kind of small scale solar absorption air-conditioning system cannot be utilized on domestic buildings because of their investment costs and competition with other alternative cooling systems. Probably, some innovation and optimisation at this sector will allow the attractiveness of the system. For example; space heating system will be integrated the system. As a result, outputs of the system show that a small scale solar absorption cooling system will be a new sector to provide environmental-friendly cooling for domestic use. Additionally, cooled water temperature was achieved approximately 7 °C that proves the system will be integrated for domestic building at this scale.

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